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NOVEMBER 7, 1911.

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POWER

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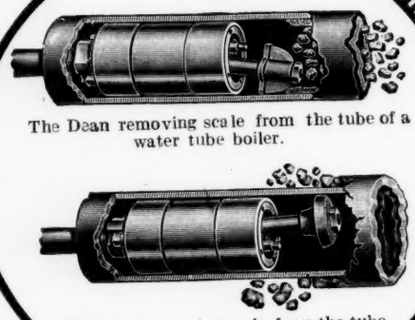

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The Dean removing scale from the tube of a return tubular boiler.

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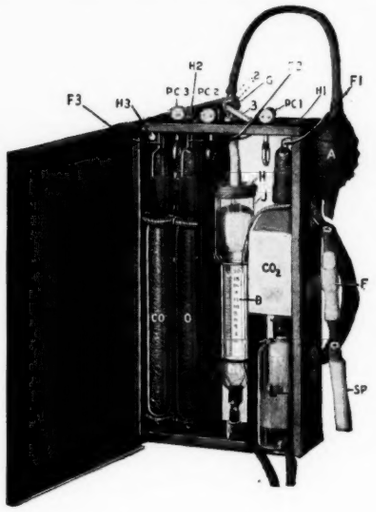

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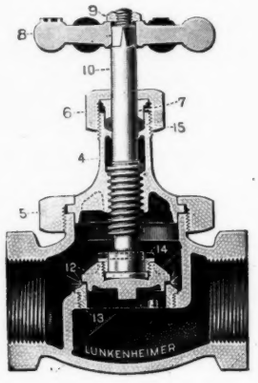

WHAT THE "RENEWO" IS

A valve that remains tight longer than any other. Should it ever leak, it can be repaired without any expense other than a slight amount of labor necessary to renew the seating surfaces. If any part (no matter what) is damaged or worn, it can be RENEWED with little labor or expense. This must be conceded by anyone having had even the Slightest experience in the use of valves as the attainment of VALVE PERFECTION, or as nearly so as is possible.

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BECAUSE it is so designed that it is practically indestructible. BECAUSE only the highest grade of bronze and nickel composition is used, insuring its durability. BECAUSE it is made by skilled mechanics in the most up to date valve factory in the world. BECAUSE the rigid inspection system controlling every item of its manufacture from the receipt of the raw materials up to the very minute the finished valve leaves the factory, insures as perfect a product as any human agency can develop. And BECAUSE it is made by THE LUNKENHEIMER COMPANY, and is the result of half a century's experience in valve design and manufacture.

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VARIOUS PATTERNS AND SIZES—The "Renewo" is made in two weights, known as the medium and extra heavy patterns. The medium pattern is guaranteed for working pressures up to 200 pounds per square inch, and the extra heavy pattern for 300 pounds. Every valve is tested under conditions more severe than occur in practice at these pressures and therefore we can safely guarantee its use. The valve is made in sizes ranging from ¼ to 3 inches inclusive.

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POWER

Vol. 34 NEW YORK, NOVEMBER 7, 1911 No. 19

A representative of a contracting firm was inspecting some repair work in one of the large school buildings of about 60 class rooms in a large Eastern city when the temperature outside was about 15 degrees above zero.

The attendant was complaining that he had to start the heating apparatus in order to have the building comfortable, Monday morning.

The representative, in great surprise, asked the janitor if he was not in the habit of operating the heating system in this kind of weather continuously, even when there were no sessions? "Oh, no," he replied, "except for banked fires, it is only in such weather as this that we run for several hours before school opens and, of course, during school sessions."

Imagine a commercial building of this size with the heating system practically inoperative for such a long period and estimate the extra boiler power required to supply sufficient steam in a few hours to replace the heat radiated through the floors and walls. This explains why there were three 54-inch boilers in the building, all operating on a gravity-return heating system with slow fires and a steam pressure of from 0 to 5 pounds.

There seems to be a great aversion to designing heating systems for school buildings that require any skill to operate, notwithstanding the fact that the horizontal tubular boiler is built precisely the same, whether to be operated at 5 pounds pressure or 100, and the fuel to be handled will be greater in proportion for the low-pressure arrangement.

It would be a large commercial building that would require boiler capacity to the above extent for the power and heating combined.

Why should large school buildings be equipped with apparatus of the house-heating type and operated in the same manner, any more than horizontal tubular boilers should be used in large public-service power stations?

Why not operate and equip large schools with the same plant arrangement as for large buildings of other types, burning the lower-priced grades of coal, and operate the boilers at their full capacity under higher pressure, with efficient attendants on duty at all times?

The enormous fresh-air supply required only during sessions is an added reason why this method would prove more economical. The higher boiler pressure enables the same boiler to carry loads of wider variation.

In the school previously referred to, large-size anthracite coal such as is used in the family furnace at $6 per ton is burned, and about twice as much boiler capacity is installed as would be needed at higher pressures. The city pays for the extra boilers and attendant repairs as well as the coal wasted in handling and forcing the extra fires.

The janitor, who is a licensed engineer, receives in some cases a lump sum for the labor; the day fireman is a cleaner most of the time and the night fireman is dispensed with altogether, at a saving of about the cost of one-third ton of coal per day.

The article on other pages in this issue shows how a school may be arranged at a lower first cost to operate with the same economy as a commercial building with the expenditure of a very few dollars for efficient help in the fire room.

A Modern Mine Power Plant

By Guy C. Stoltz* and Samuel Shapira†

The new power plant of Witherbee, Sherman & Co., located at Port Henry, N. Y., is equipped with 1864 horsepower of water-tube boilers provided with superheaters, turbine blowers and damper regulators. The main units are four-stage Curtis turbo-generators.

*Chief engineer, Witherbee, Sherman & Co., Mineville, N. Y.
†Mining engineer, Witherbee, Sherman & Co., Mineville, N. Y.

Witherbee, Sherman & Co., operating magnetite mines at Mineville, N. Y., are equipped to produce over 1,000,000 tons of high-grade ore annually. A small percentage of the output is shipped without milling, averaging about 60 per cent. iron and 1.3 per cent. phosphorus, while a greater proportion of the ore hoisted is concentrated electromagnetically, the iron content being raised to 65 per cent. and the phosphorus reduced to 0.75 per cent. Special high-grade concentrates can, if required, be milled to 71 per cent. iron and 0.17 per cent. phosphorus.

To operate economically the several mines, each so located that individual hoists, compressors and concentrating plants are necessary, it was deemed expedient to install new power plants and electrify the entire surface and underground equipment. With entire electrification in view, the company has built a two-unit turbo-generator plant, 800 and 1500 kilowatts respectively, at Port Henry, N. Y., six miles from the mines, and has further installed at a Corliss engine, 750-kilowatt generator plant at Mineville, a 750-kilowatt, low-pressure turbo-generator set. Two hydroelectric plants at Wadhams Mills and Kingdom can be counted upon during a greater part of the year for a 675-kilowatt combined load.

The Port Henry plant embodies the latest practice in a strictly steam-turbine plant operating condensing and using superheated steam. The power equipment is noteworthy because of its magnitude and its thoroughly modern design. The electrical installation at all plants was furnished by the General Electric Company, and was under the direction of H. Comstock, assistant general manager, and H. F. Pigg, electrical engineer.

The building, roughly T-shaped, of concrete and steel construction, was designed by H. C. Pelton, of New York. The floor space, 7500 square feet, includes 530 square feet of mezzanine floors. Two-thirds of the area is occupied by the boilers and the remainder by turbines and auxiliaries. The walls are 49 feet high to the coping, and are windowed to give a generous supply of natural light. The cubic contents of the building are 373,173 cubic feet, and it was erected at a cost of approximately $87,500, or 23½ cents per cubic foot.

Handling of Coal

No. 2 buckwheat anthracite coal is used as fuel, of which 11 tons, with proper draft and fired on suitable grates, will equal the steaming value of 10 tons of bituminous. The total cost of a ton of this fuel delivered to the plant is $2.65 (which includes 15 cents per ton for unloading) while the cost for bituminous is $3.92 per ton. The coal gondolas from the Delaware & Hudson main line are pushed up an eighth-mile spur and dumped between the concrete bents of a trestle 100 feet long, on the north side of the building and 25 feet distant from the front of the boilers.

Fig. 1. Port Henry, N. Y., Power Plant

At present no mechanical fuel-handling system has been installed; several methods, however, are being considered. All coal is now being handled in wheelbarrows of 400 pounds capacity, which pass over scales to the boilers. Weighing is necessary in order to determine the daily fuel cost per kilowatt-hour so that an intelligent charge may be made at the station for power distributed from this plant.

The Stack

Natural and forced draft are required. The former is obtained by a circular, reinforced, monolithic, concrete stack. The stack is situated centrally between the line of boiler sets, stands 175 feet high, and has an internal diameter of 8 feet. A 4-inch air space is provided in the lower 47 feet of the stack, the outside wall being 7 inches thick and reinforced longitudinally with 1¼x1¼x⅜-inch T-bars; the inside wall is 4 inches thick and is provided with the same reinforcement, 16 lengths in all being used, while the horizontal ⅝-inch round iron bars are spaced on 18-inch centers. The outside shell is reinforced with 80 vertical bars, the number diminishing to 32 in the interval of 65 to 85 feet above the surface. The upper 90 feet, which is a single shell 5 inches thick, has 16 vertical bars, together with the ⅝-inch iron bars on 18-inch centers, equivalent to a reinforcement of 0.46 per cent. Around the 3x8-foot breeching entrance to a hight of 12 feet the stack is lined with firebrick.

Boiler Room

The boiler room, occupying by far the greater part of the building, is 105 feet long by 48 feet wide by 49 feet high. The floor is of concrete and slopes from the side walls to the pipe trenches.

The equipment consists of six sets of Babcock & Wilcox water-tube boilers with a combined capacity of 1864 horsepower. Four sets of boilers are rated at 266 horsepower each and the other two sets at 400 horsepower each, all being equipped with superheaters. The floor space occupied by this installation is 1745 square feet, or 35 per cent. of the total boiler-room area. All the boilers are provided with No. 3 McClave grates, having ⅜-inch mesh openings. The grates are especially adapted to burn the smaller sizes of anthracite, and are so made that each row of grate bars is divided into a front and rear series by means of two separate connecting bars, operated by twin stub levers and connecting rods, with an operating handle adapted to grasp either one or both of the levers in such a manner that the front and rear series may be operated separately or together. In the shaking movement of the grates there is no increase in the size of openings. The grates have an area of 65 square feet, and a slope of ½ inch per foot.

Blowers

Each boiler is equipped with Wing turbine blowers to provide forced draft. The 400-horsepower boilers are provided with two 20-inch blowers, while each of the 266-horsepower boilers has a 20-

inch blower. With the aid of the blowers, 25 per cent. increase in boiler capacity is obtained while using any fair grade of fuel. Being operated by a balanced valve, through a damper regulator, the blower is in operation only on a fall of steam pressure, thus keeping the pressure constant within a few pounds. The blower is installed in the side wall of the boiler, and thus requires no floor space or attention aside from oiling.

A Davis damper regulator is used in connection with the Wing blowers. All the blowers and main dampers of the smoke breeching are connected to this automatic regulator, which is guaranteed to maintain a pressure within 2 pounds of a fixed amount, 160 pounds in this case.

Pumps

Two main pumps, 10 and 6 by 10-inch Knowles, and one large Knowles 16 and 5 by 12-inch compound, duplex, pot-valve pump are used to supply the necessary water to the boilers. Under ordinary working conditions, the two smaller pumps are sufficient, the larger pump being held in reserve. The water comes from the condensers in the turbine room to a large tank 20 feet diameter and 18 feet high, open at the top, which serves as a storage tank and allows an escape of the air which comes with the condensed water. The tank is located directly in front of the stack in the boiler room and is supported on steel legs resting on a reinforced-concrete platform 10½ feet above the boiler-room floor. The water from this tank is divided to flow to two Cochrane 1000-horsepower feed-water heaters, each having a 5-inch suction outlet, a 10-inch exhaust outlet, a 10-inch exhaust inlet and a 3½-inch waste pipe, which also rests on the platform. The Cochrane heaters receive exhaust steam from all the auxiliary pumps to heat the feed water. The remainder of the feed water is supplied by two small Knowles pumps. All pumps are tied into the same supply system and a 4-inch pipe line is used for boiler feed.

Piping

Superheated steam from each of the boilers is taken through a 7-inch, extra-heavy pipe, inverted U-shaped, with 12-foot spread, and delivered to a 10-inch extra-heavy collecting header placed in the rear of the boilers and extending the entire length of the room. From this main header, two 7-inch, extra-heavy inverted-U pipes take the superheated steam to the turbine. This header is equipped with five 10-inch, extra-heavy bypass gate valves so placed that the steam may be taken from any set of boilers or all, according to the need of the plant. Running parallel with the main superheated-steam header is a 2½-inch auxiliary saturated steam header, from which proper leads are taken to the auxiliaries. Extra-heavy piping and fittings are used on all the superheated lines, and heavy piping and fittings on all other steam lines. The superheated-steam pipes are covered with 85 per cent. magnesia blocks 1 inch thick and 3 inches wide, and with double standard thickness

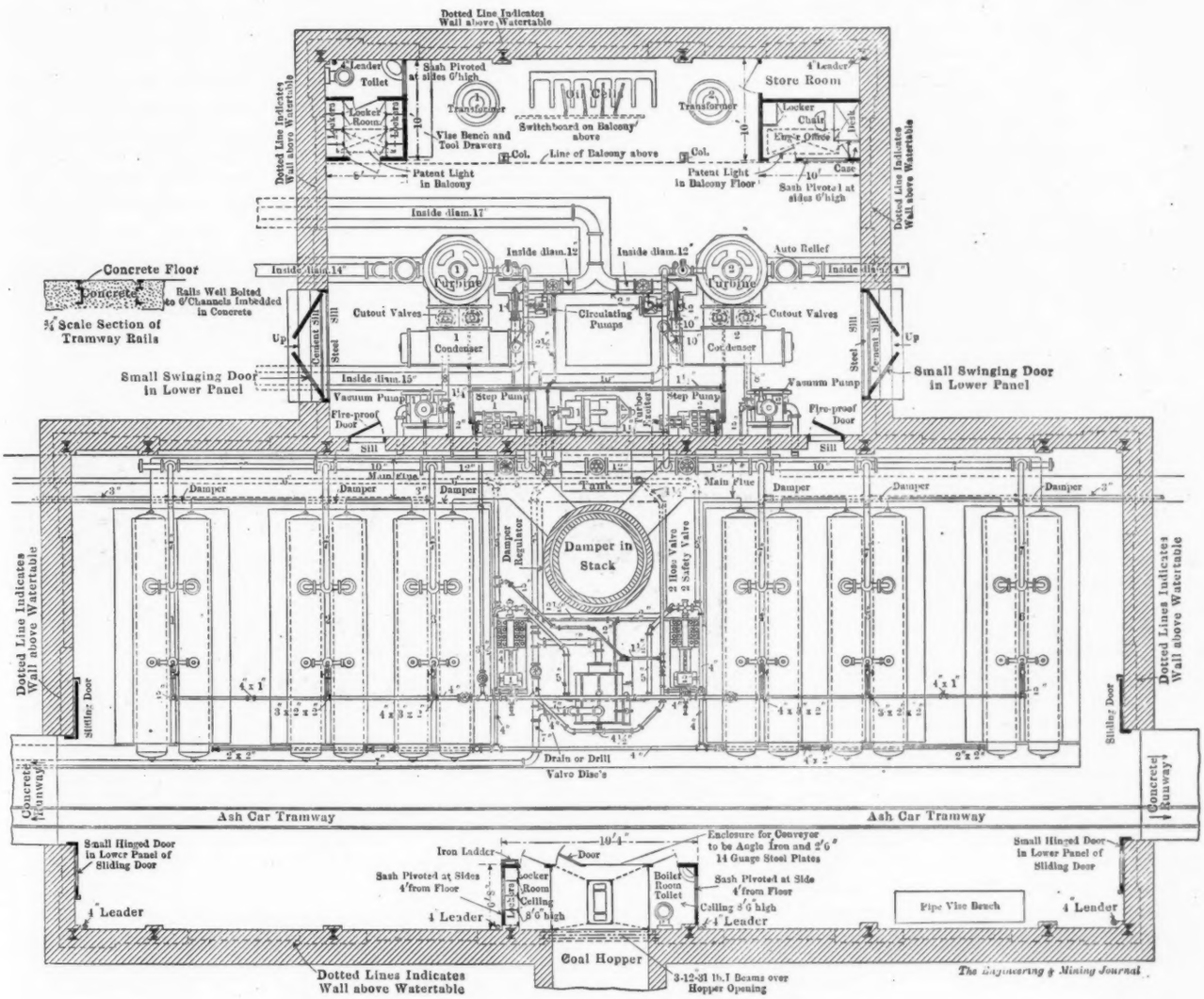


FIG. 2. LAYOUT OF THE WITHERBEE-SHERMAN POWER PLANT

sectional covering; the outer layer is of asbestos sponge felt, and the total thickness of the covering is 3½ inches. The covering for the other steam pipes is sectional asbestos fire felt of standard thickness.

Smoke

The 4x9-foot smoke breeching is made up of No. 10 gage iron plate laid on a framework of 2x2x1¼-inch angles. It is supported by iron brackets from the partition wall and is furnished with two main dampers, one on each side of the stack and near the stack openings. These dampers are so hung on ball bearings that there is the smallest amount of friction, and they are free to open and close when worked by the Davis regulator, to which, by means of suitable cords and pulleys, the damper arms are attached. Besides these two main dampers, each boiler is provided with hand-operated dampers placed at the boiler-flue openings and breeching connections.

At present, ashes are handled with wheelbarrows and before dumping they are weighed. A McClave-Brooks ash-conveyer system is to be installed, and this, with the proposed coal-handling system, will make the boiler room uptodate in every respect.

Turbines and Auxiliaries

The turbine room, 38x52 feet, contains two Curtis four-stage turbo-generators, one of 800 kilowatts, 6600 volts, running at 1500 revolutions per minute and a second of 150 kilowatts and the same speed and voltage. Both run on oil-step bearings and are controlled by hydraulically operated governors. The turbines are placed on solid blocks of concrete standing 7 and 9 feet respectively, above the main-floor level. Stairs lead to an iron-grate walkway which surrounds both turbines and extends to the north partition wall where are situated the oil pumps, separators and other auxiliaries. A full-load water rate of 18 pounds per kilowatt-hour is guaranteed on the larger unit and a rate of 18.2 pounds is given for the smaller turbine, running condensing. Steam at 160 pounds pressure and superheated 100 degrees is delivered to the turbines, while all of the auxiliaries operate on saturated steam.

Excitation for the turbo-generators is furnished by a 35-kilowatt induction motor-generator set, and by a 25-kilowatt horizontal single-stage turbo-generator; either set has ample capacity to supply excitation for both generators.

The condensing apparatus for the turbines is of the surface type, and consists of two circulating pumps and two air pumps supplied by the Wheeler Condenser Engineering Company. A vacuum of 28 inches is maintained. The centrifugal circulating pumps are direct connected to vertical high-speed engines. The air pumps are of the Edwards vertical type, 8 and 20 by 12 inches, and deliver the condensed steam to a standpipe and finally to Cochrane heaters of the open type.

The oil from the step bearings runs to a collecting tank of 100 gallons capacity, from which it is delivered by a Knowles pump to a Niles filter of 600 gallons capacity, where oil and water separation is effected. If necessary, the oil can be heated to lessen its density before filtering. The filtered oil is then delivered to one of two step pumps; both are Dean, 4 and 2½ by 6-inch, outside-packed plunger type, and operate at 350 and 425 pounds pressure respectively. The oil from the tank is delivered to an overhead reservoir at 130 pounds pressure by two Blake 3x4-inch duplex oil pumps, piston type. From the receiver the oil flows to the hydraulic governor and to the top and middle bearings of the main turbines through bafflers.

The larger turbine is cooled by forced ventilation effected by a disk fan which is mounted on the main shaft. The stator and rotor on the smaller unit are cooled by a fan circulation piped to the generator.

A 20-kilowatt horizontal single-stage turbo-generator set provides for lighting the building and yards. A mezzanine floor 9 feet above the main floor is built against the south wall of the room to receive the switchboard. One of the switchboard panels is for the exciter, one for the regulator, two for the generators and two for the lines. The cells for oil switches, transformers for motor-generator sets, exciters and all high-potential wiring are located under the switchboard floor. All piping, where possible, is distributed to the units through concrete trenches in the floor, the trenches being covered with removable cast-iron plates. One 30-ton Brownhoist crane is installed in the turbine room. The entire equipment in this room can be operated by one engineer and one oiler for each shift. The entire cost of equipment at the plant was approximately $160,000 installed.

Power Transmission

Three-phase current at 6600 volts is sent from the plant over a double transmission line, two sets of three wires each No. 000 and No. 00 Brown & Sharpe gage. The lines are strung on wooden poles fitted with standard crossarms, provided with locust pins and porcelain insulators. All highway crossings are protected by safety aprons strung from pole to pole. Leaving the main building, the transmission lines run through a small building of monolithic concrete which contains lightning arresters of the shunted multigap type.

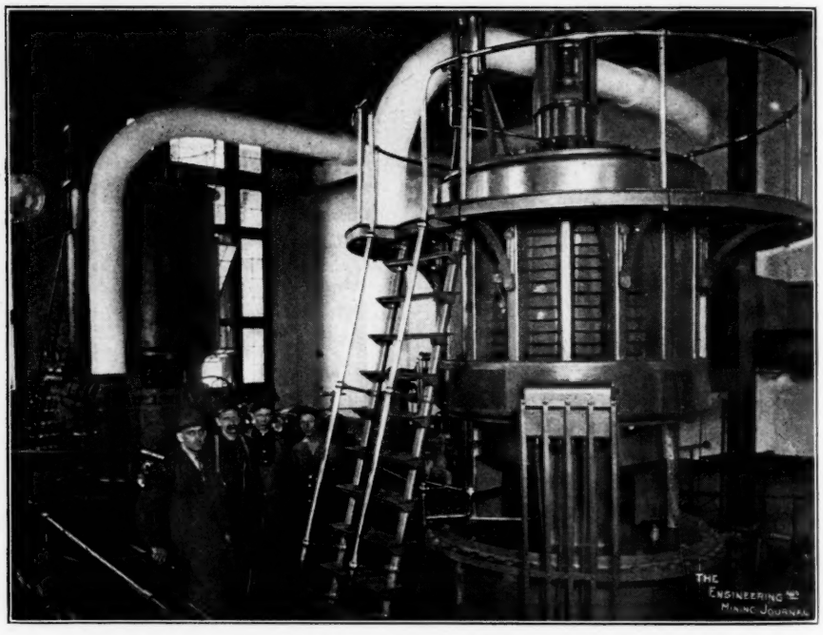


Fig. 3. Turbines at the Witherbee-Sherman Plant

Transforming Station

At Mineville, before entering the transformer station, the high-potential lines run through the A & B distribution station, in which is installed a second set of arresters of the multigap type. Beyond the arresters and disconnecting knife switches the lines are carried through automatic overload switches and then to the main transformer station.

The station, of red brick and monolithic concrete, is 20x20 feet. It contains three 500-kilowatt, 6600-3300 air-cooled transformers, of the same type; also four Sturtevant fans directly driven by 2-horsepower, 110-volt induction motors for supplying the cooling air. From this station the lines are taken to the main distribution station located in the A & B hoist and compressor house.

Pulleys for High Speed Belts

By Karl W. Knorr

Slippage with high belt speeds is not due to an air cushion being drawn between belt and pulley but to the action of centrifugal force which stretches the belt and tends to make it rise off the pulley face.

It is well known among those who have had experience in operating belts that when the belt speed is high, difficulty is encountered in maintaining the proper speed ratio between the driving and driven pulleys. The speed of the driven pulley falls below what it should be according to the figures obtained from the formula commonly used for figuring pulley speeds.

As the correct operation of any machine is dependent upon its running up to speed, this lagging of the driven pulley behind its figured speed causes considerable annoyance, to say nothing of loss in machine efficiency. It is particularly objectionable because there seems to be no rule by which just the proper allowance for lagging can be made, thus bringing the machine to the right speed, notwithstanding the slippage. Then, there is, of course, the matter of lost power to consider.

There has been more or less difference of opinion regarding the cause or causes of this slippage, for so it must be termed, and a number of theories have been advanced in explanation. The one which has gained the most general credence is based upon the observation of the fact that when a belt operates at a high speed it seems to leave the driven pulley so that it is often possible to see between the belt and the pulley.

According to this theory the high speed of the belt causes air to be drawn in under the belt, forming an air cushion between the pulley and the belt, upon which the belt rides, skimming around the pulley instead of coming into close contact with its face. This theory holds that the rotation of the pulley is due to the friction between this revolving air cushion and the face of the pulley.

If this theory is correct, obviously the solution of the whole trouble is to do away with the air cushion which prevents the belt from coming into close contact with the face of the pulley.

It would seem that the easiest way to accomplish this would be the introduction of a small deflecting shield in front of the pulley, and so placed that the belt would just pass without touching it. If any air current existed this deflector would surely break it up and prevent its going under the belt, which it does according to the theory. If this method has ever been suggested, it does not seem to have gained any foothold.

The most popular method of doing away with this so called air cushion is the use of pulleys the rims of which are perforated to permit the air to pass through, or grooved circumferentially to permit the air passing out under the belt. A number of manufacturers make and exploit these perforated and grooved pulleys, but they do so probably because there is a demand for them rather than because they believe there is any real merit in the idea.

For a cushion of air to force itself between a belt and the pulley over which this belt operates, it must have sufficient pressure to overcome the pressure of the belt on the pulley produced by the initial belt tension and the load tension.

In the character of service where high belt speeds are more generally used, viz., motor and generator work, both the initial and the load tensions are, as a rule, very high, so that the air must be under considerable pressure to lift the belt from the pulley.

For illustration, take a 10-inch belt running over a 16-inch pulley. An initial tension of 40 pounds and a load tension of 60 pounds per inch in width of belt is a modest assumption, for in practice the figures will run nearer 60 pounds initial tension and from 90 to 120 pounds load tension. However, for illustration, it is desirable that the figures be conservative rather than high. Taking the figures assumed, it will be seen that the total pressure on the pulley per inch in width of belt is 140 pounds, this being the sum of the 40 pounds initial, the 60 pounds load tension on the tight side of the drive, and 40 pounds initial tension on the slack side of the drive. The belt is 10 inches wide, so that the total pressure of the belt upon the face of the pulley is 1400 pounds, and as the projected area of the pulley is 160 square inches, the pressure of the belt per square inch of projected area is 8¾ pounds.

In order to form a cushion under the belt, the air must not only overcome this heavy pressure, but it must also have sufficient extra pressure to produce the added tension necessary in the belt to straighten out the catenary curve in which it hangs, and by thus reducing the sag permit the belt to ride on the surface of the cushion, or the extra pressure must be sufficient to stretch the belt enough to allow it to ride on the cushion. The cushion of air has the effect of increasing the diameter of the pulley so far as the belt is concerned, and its presence, therefore, necessitates an increase in the tension of the belt, reducing its sag between the pulleys, or causing a stretching of the belt.

The tension necessary to produce either of the results mentioned is considerable, so it will be readily seen that the pressure of the air forming the cushion must be extremely high.

If the belt will produce a current of air on its under side which will attain the extremely high pressure necessary to produce the results claimed, why will there not be a similar stratum of air on the other side of the belt? That this is not the case is shown by the absence of results which would accompany a stream of air at from 9 to 12 pounds pressure per square inch, and of a width equivalent to that of the belt. One would not have to get very close to such a stream to be made fully aware of its existence.

It would seem fair to assume that if the current of air under high pressure does not exist on the outside of the belt it does not exist on the inside; but even if it does, it is difficult to conceive why this current will not rush out sideways when it strikes the face of the pulley, taking the line of least resistance instead of overcoming all the resistance offered by the belt and going under it.

Then, too, if the air does go under the belt, why will it run through holes or grooves made in the rim and not out the sides of the belt?

Another point which seems strange is that the cushion always forms on the driven pulley. If the current of air exists, would it not perform in the same way on the driving pulley as on the driven?

A consideration of the effects of centrifugal force develops what would seem to be a rational explanation of the phenomenon. The stretching of the belt is due to centrifugal tension, the belt coming in contact with the driven pulley at frequent intervals, thus driving it, but jumping away frequently enough to produce the illusion of running on air.

There is no question of the existence of such a tension to figure the amount of which Rankine gives the following rule: If an endless band, of any figure whatsoever, runs at a given speed, the centrifugal force produces a uniform tension at each cross-section of the band, equal to the weight of a piece of the band, whose length is twice the hight from which a heavy body must fall, in order to acquire the velocity of the band.

In the case of a leather belt, this ten-

sion produced by centrifugal force can be expressed by the formula

$$T = 0.012\ V^2,$$

where T equals the centrifugal tension per inch in width of belt, and V equals the velocity of the belt in feet per second.

Using this tension and a fair average value of the coefficient of elasticity of leather, the formula for the stretch of leather belting assumes the form

$$S = 0.00000049\ L\ V^2,$$

where

S = Stretch in inches,
L = Total length of belt in inches,
V = Velocity of belt in feet per second.

That the trouble is due to the stretching of the belt produced by centrifugal tension is borne out by the following:

The lower the belt speed the lower will be the centrifugal tension, and, consequently, the less will be the belt stretch. It is, therefore, necessary that the belt be kept up to speed if there is to be any stretch due to centrifugal tension so the belt must necessarily hug the driver closely and the stretch, if any, must all be evident at the opposite end of the drive. This agrees with observed facts.

As the centrifugal tension is reduced with the speed, it is probable that there should be a speed below which the tension will not be sufficient to produce objectionable results. It has been found in practice that if belts are run at speeds not exceeding 4500 to 4800 feet per minute, no trouble is experienced on account of undue slippage.

If centrifugal tension causes slippage, the clearance between the driven pulley and the belt should increase with the speed of the belt. This has been observed to be the case in practice.

Also, if this trouble is due to increased length of belt due to centrifugal tension, it should be possible, after getting the belt up to speed, to eliminate the clearance between the belt and the driven wheel by backing the latter up to the position maintained by the belt. This is also true in practice. Of course, when the belt is stopped and the centrifugal tension eliminated, the driven wheel must be pulled forward again. When the belt is in motion the centrifugal tension stretches it, and when the belt is stopped it tends to regain its original length. If the driven wheel is not pushed forward when the belt is stopped, the elasticity of the belt tending to bring it back to its normal length will produce an enormous pressure on the bearings, and the belt will be under excessive internal strain while at rest. With generators and motors it is no great trouble to run the pulleys back into the belts, or pull the belts into the pulleys, because they are mounted in sliding frames with adjusting screws.

The foregoing is without doubt the real solution of belt slippage with high-speed leather belts.

A means of overcoming the trouble and at the same time doing away with the necessity of moving the pulleys would be to use a belt made of some material which does not stretch as easily as leather. A steel-band belt is being exploited to some extent at the present time, and considerable success in operation is being claimed for it. As steel will stretch only about one-thousandth as much as leather under the same tension, it is obvious that a belt made of this material would be better for high-speed service than one made of leather.

Firing Marine Boilers on the Chicago River

Instructions for firing marine boilers in vessels without making dense smoke while in the Chicago river and harbor, which apply to the average freight and passenger boat, formulated by Osborn Monnett, smoke inspector, and Edwin F. Oyster, marine deputy, have been issued by the department of smoke inspection of the city of Chicago. While the advice is given as specifically as possible, yet there are instances when a little deviation might bring better results. What may apply to one boat might not apply to others, but the instructions are so general that freight and passenger boats having either Scotch marine or firebox boilers can use them to advantage in the effort to eliminate dense smoke. The instructions are as follows:

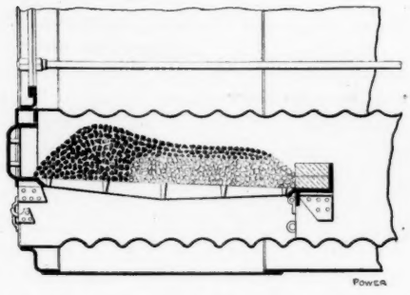


FIG. 1. FIRES IN CONDITION TO ENTER CHICAGO HARBOR

COMING INTO PORT

Clean all fires before reaching Chicago.

Before entering the harbor, shove the fuel to the rear of the grates, clearing about 2 feet of the front of the grate surface; cover the burning fuel with about 4 inches of fresh coal and fill the cleared space with broken coal almost to the top of the furnace; leave door cracked for short interval and put on the blower. Have all fires in the condition shown in Fig. 1 when abreast the life-saving station. The vessel will then be able to reach her dock without again working the fires.

All further firing while in the Chicago river should be with Pocahontas coal.

FIRING AT THE DOCK

When steam is required, or the fire on the rear of the grate gets low, push the coked coal back and quickly charge the front of furnace with fresh coal. Leave the door cracked for short time and keep the blower on until the top of the fire assumes a bright red color. Regulate the steam pressure by the damper. Fig. 2 illustrates the condition of the fire before pushing it back.

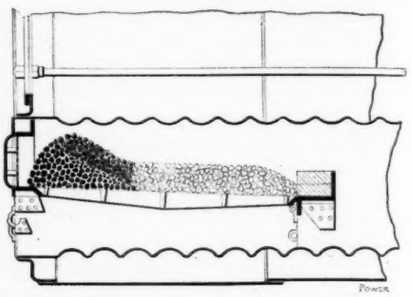


FIG. 2. FIRE READY TO BE PUSHED BACK

CLEANING FIRES IN PORT

Avoid, if possible, the cleaning of fires in port. If necessary to do so, clean quickly and when convenient only one boiler during an hour.

Throw broken coal in front wings of the furnace on each side for a distance of 3 or 4 feet. Crack the door and use the blower. Wait until the last charge of coal is burning well, close the ashpit door; rake out in front all the old fuel on the central portion of the grate and fill the cleared space with well broken-up coal to a thickness of 4 or 5 inches. If steam is not required, let the green fuel catch and get well coked from the live beds at the sides.

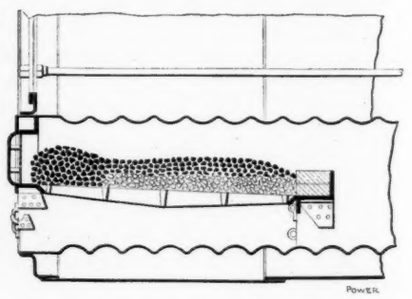


FIG. 3. SHOWING FIRE AFTER COKED COAL HAS BEEN PUSHED BACK

To raise steam more quickly, the burning fuel at the sides can be winged over on top of the green coal. After either operation, open the ashpit door and put on the blower until the fire gets action.

After the middle portion has ignited well, rake out the sides, which will be nearly burned out, charge with fresh

broken-up coal and allow this to burn without winging over the center. Crack the doors and use the blower.

Fig. 5 shows the condition of the fire when cleaning by this method.

Give plenty of time for the fresh coal to coke before again disturbing the fires.

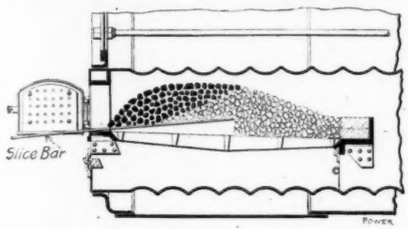


FIG. 4. METHOD OF BARRING FIRES

Do all the cleaning and charging as quickly as possible in order to preserve the furnace temperature.

WHILE SHIFTING IN RIVER

A half-hour's notice or more should be given the fireman before getting under way in order to have steam enough to shift without forcing the fires.

Raise steam by stirring the top part of the fire with a rake and breaking up the coked portion of the fuel bed; crack the door and use the blower.

Allow the fire to burn brightly for a few minutes; push back and put new charge on the front of the grate. Repeat as often as necessary to hold steam.

To make additional steam in shifting, fuel beds may be barred if necessary, but this should consist only of a slight raising of the bed and not passing the bar up through it, as clinkers will mix with the fuel, spoiling the fire and making smoke. See Fig. 4.

Lumps of fuel charged should never be larger than one's fist.

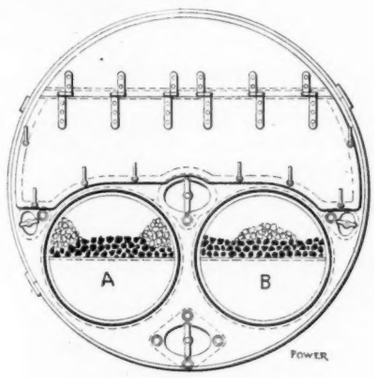


FIG. 5

A, central portion of grate after being cleaned out and charged with fresh coal. *B*, wings of furnace cleaned and charged with fresh coal after central portion of fuel bed has ignited.

Cracking of the doors consists in the doors being opened not over 1 inch and as they are opened for short periods of time only, the flues will not be affected.

While in the river, fires must be carried very thick at all times. If the fires are allowed to burn down to 3 or 4 inches at the dock, it will be impossible to make steam without making smoke.

STARTING FIRES

Cover the bare grate with well broken-up coal to an average thickness of about 4 inches. Then put five scoops of live coal on the top at the bridgewall as shown in Fig. 6, and close the fire door with the damper open. Fire will burn from the back to the front with much less smoke by this method than by any other.

When the fire has covered the entire grate, push the fuel bed toward the rear and fill the front of the grate with fresh coal to about one-half the hight of the furnace. This should be left undisturbed until mostly coked. Crack the door and use the blower. Shove the coked coal back and repeat the operation as often as it is necessary to raise steam.

In all operations where the fire is disturbed, give the fire a little air through the doors and use the blower.

BURNING POCAHONTAS COAL

This fuel cakes quite easily and to get the required steam it must be broken

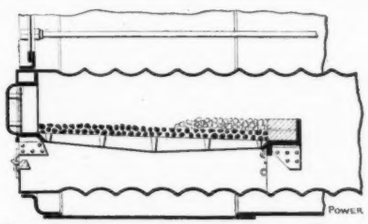


FIG. 6. METHOD OF STARTING FIRES

up occasionally from the top; use a rake or a hoe.

BLOWING FLUES

Flues should never be blown in the Chicago river.

A Problem in Statics

Recently a query was received as to the solution of the following problem:

Three men carry a stick of timber, two taking hold at a common point and the third at the end. Where should the two men take hold together so that the load will be equally distributed among the three?

The problem is an old one but it still continues to puzzle many. As its solution embodies the fundamental treatment of all beams, however, it is herewith given.

FIG. 1. TIMBER BALANCED

If a man supported the beam under its center of gravity, it would balance on his shoulder and he would carry the whole weight, as in the sketch, Fig. 1. A man standing at one end would bear none of it. However, as the middle man moved toward the other end more and more of the weight would come upon the end man, until when each was at an equal distance from the center of gravity the load would be equally divided, as in Fig. 2.

FIG. 2. EACH MAN SUPPORTING HALF THE WEIGHT

In all cases the product of the load carried and the distance of the point of support from the center of gravity will be constant. As it is stipulated that the end man shall carry one-third the load W, and as his distance from the center of gravity is half the length of the timber, or $\frac{L}{2}$, the product of this distance and his share of the load will be

$$\frac{L}{2} \times \frac{W}{3} = \frac{WL}{6}$$

The other two men are to carry two-thirds of the load together. Letting their distance from the center of gravity be represented by the unknown value x, the product of this distance and their load must be

$$\frac{2W}{3} \times x$$

But since the product of the load carried by a support and its distance from the center of gravity is constant for any given case, then $\frac{WL}{6}$ must equal $\frac{2Wx}{3}$. From which

$$x = \frac{WL}{6} \div \frac{2W}{3} = \frac{L}{4}$$

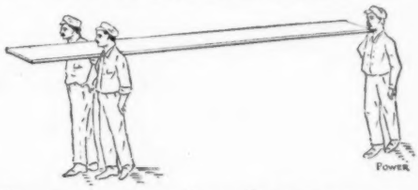


FIG. 3. LOAD EQUALLY DISTRIBUTED BETWEEN THE THREE MEN

Hence the two men support the timber at one-quarter its length from the middle; in other words, three-quarters of the length from the end supported by the single man, as shown in the illustration, Fig. 3.

It is claimed that the first steam derrick was used by James J. Smith in building the New York post office foundations, in 1870. Knowing their wide use today, how did we ever get along without them up to this late date?

Superheating and the Superheater

By L. B. Taylor

Superheat reduces radiation loss and improves the heat utilization in the engine. Another advantage is reduction in necessary boiler and condenser capacity with consequent reduction in feed and condensing water. A formula is given for estimating the superheating surface required for a given degree of superheat and a given set of conditions.

Recent tests on certain high-grade German types of combined boiler and engine, with about 200 degrees Fahrenheit of superheat, showed as high as 25 per cent. increase in plant power, 25 per cent. saved in coal and 33 per cent. saved in boiler-feed water, due to the superheating. For the ordinary superheats used, in Europe at least, an average saving of 7 per cent. in coal may be assumed.

Lightening the Work of the Boiler

For new installations this saving in coal ought to be effected without much, if any, extra cost of the boiler plant, for the cost of a superheated steam plant is approximately equal to that of a saturated steam plant of the same power, since the superheated plant has less steam to generate. This will be clear by referring to Fig. 1, which shows the relative prices of water-tube boilers and their superheaters for different amounts of boiler-heating surface. The superheaters for large boilers cost from 20 to 25 per cent. as much as the boiler itself, so that if by the use of the superheater the power of the plant is increased in like proportion to this extra cost, the superheated plant will cost no more than the saturated plant of the same power.

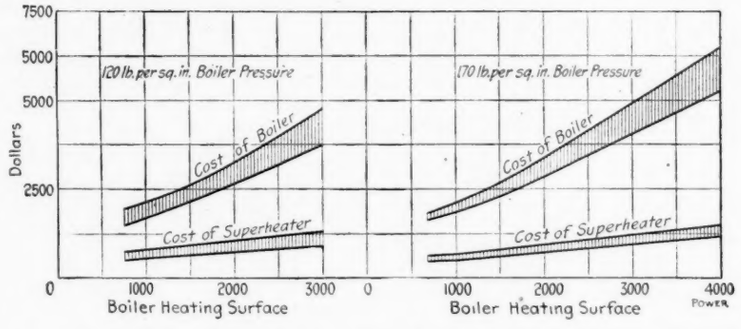


Fig. 1. Approximate Cost (in Germany) of Boilers and Superheaters

Although the advantage of superheating is principally due to the engine or turbine operating on the superheat, under certain conditions there is also an improvement in the heat utilization in the boiler, for when saturated steam is heated at constant pressure its volume increases, hence a boiler with the same amount of boiler-heating surface will generate more superheated steam, measured by volume, than saturated steam. It is a common error, however, to assume that with superheating a better heat utilization in the boiler is obtained under all conditions, for this is not always so. The better heat utilization depends greatly on the boiler load and is sometimes very small. This should be considered in determining the size of the boiler. A boiler for superheated steam may be smaller than one for saturated, to give the same power, which is another point in favor of low plant cost.

Since the water space in a boiler is a measure of the energy reserve, the relative disadvantage of the small volume of water in the water-tube boiler is diminished by the employment of a superheater, for this helps to store energy. Internally fired boilers with their large water space have less to gain in this respect. Superheaters are useful with water-tube boilers in drying out the moist steam generated, since this type of boiler has a relatively small steam space which is the measure for the dryness of the steam. Another advantage of superheating is the elimination of loss due to condensation in the piping.

The reduction of the pipe loss and the thermal loss in the engine, which is the same as increasing the thermal efficiency, is obtained because the loss due to cooling until the steam reaches the saturation limit is taken from the superheat and not from the latent heat; hence no condensation is caused. This is clearly the principal economic reason for using superheat. Superheated steam has the same economical importance for long steam-pipe lines as it has for the prime mover. By using high pressures and high superheats, and then by stepping down with reducing valves at destination, small pipes and high velocities (twice that for saturated steam) can be used, resulting in a low first cost and in small heat losses, so that transmission of steam for distances of from 6000 to 10,000 feet has recently been made possible.

Increasing Thermal Efficiency of the Steam Engine

Superheating, besides increasing the volume of the steam, results in an improved steam consumption in the engine, especially at fractional loads. This improvement is explained in Schule's "Thermodynamics" as follows:

In a steam engine working on the ideal process, by the use of superheat a somewhat larger part of the total heat supplied by the steam is converted into work, for the same initial and final pressures and the same back pressure. For steam pressures between 90 and 120 pounds per square inch, the difference in the thermal efficiency is from about 6 to about 4 per cent. in favor of the superheated steam. With increasing steam pressures this difference decreases still more. Such slight saving would in general be too small to make it worth while to install a superheater and to warrant the expense necessary for operating on superheated steam.

The actual improvement is, however, considerably greater than would be expected from theoretical reasoning. This is true because of the much smaller losses due to the cooling of the superheated steam as compared with saturated steam. The superheated steam loses heat to the cylinder walls during admission, but so long as the whole superheat is not lost, no moisture formation occurs. The steam contracts during the cooling down, but the loss in work produced thereby is much smaller than if a part of the steam, as with saturated steam, went over into the liquid state and thereby entirely lost its capacity for work. The much smaller heat conductivity of the more gaseous superheated steam, compared to the moisture-laden saturated steam, results in a decrease in the cooling-down losses. Consequently, with otherwise similar conditions, the operation with superheated steam gives a better engine efficiency and, in general, a higher commercial efficiency. Even a moderate superheat, which can be used in an engine originally built for saturated steam, may produce such an important saving that it will warrant the installation of superheaters in a plant already existing.

In practical operation a steam cylinder of a given size with a given point of

cutoff and a given initial pressure will develop less power with superheated steam than with saturated steam because the expansion line for superheated steam falls more rapidly. This fact does not conflict, however, with the fact that better utilization of the heat is obtained with superheated steam.

To overcome the effects of cylinder condensation, steam jackets, requiring a weight of steam amounting to from 10 to 12 per cent. of the operating steam, are used when running on saturated steam. Only low-speed engines are improved by these jackets. With medium and high superheat the jacket is not regularly required and is used only for starting up, as otherwise the lubrication of the piston guiding surface is endangered.

Cylinder and pipe condensation is avoided because of the fact that superheated steam is a poor conductor of heat; the coefficient of heat transmission at 100 feet per second steam velocity amounts, on the average, to only 1/12 of that of saturated steam. Furthermore, superheated steam can give up a part of its heat (from Fig. 2) for a superheat of 650 degrees, about 10 per cent. before it becomes saturated and begins to

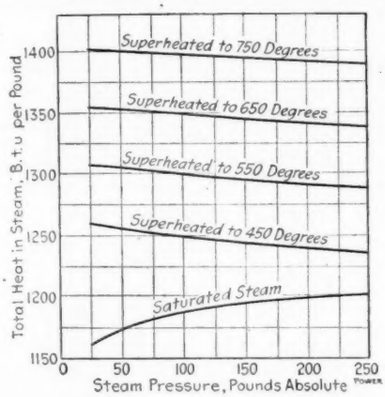


FIG. 2. TOTAL HEAT IN SATURATED AND SUPERHEATED STEAM

deposit as moisture on the cylinder or pipe walls.

An experiment proving how poor a conductor of heat superheated steam really is can be made with a small copper flask the bottom of which is first heated red hot over a bunsen burner. The flame is then removed and a test tube full of water is poured in and the opening closed lightly with a cork. It will be a surprisingly long time before the layer of superheated steam immediately formed between the water and the flask will allow enough heat to pass to boil the water, which finally flashes completely into steam, blowing out the cork with considerable force.

The reduction in the quantity of cooling water required with condensing engines and turbines, made possible by the improved steam consumption, means much if the cooling water is difficult to obtain. It is true that the steam on leaving the engine or the turbine will be drier (up to perhaps 8 per cent.) if originally superheated, but the quantity of steam used will be reduced in a considerably greater ratio, so that the total quantity of heat to be abstracted by the cooling water, and consequently, the amount of cooling water required, will be considerably reduced.

Since a drop of water acts like a solid body on the steam-turbine blading when entering with the steam, super-

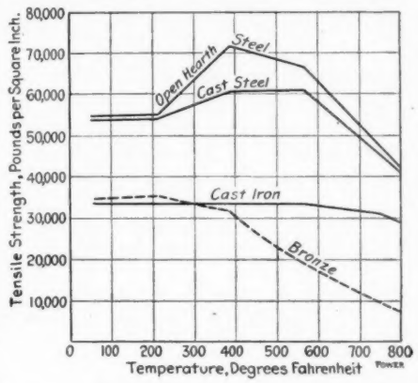


FIG. 3. EFFECT OF TEMPERATURE ON TENSILE STRENGTH OF VARIOUS METALS

heating, by avoiding the moisture, prevents erosion of the blades. Turbines running on superheat have been operated for as much as 25,000 hours without showing any noticeable wear or any increase in steam consumption.

Depending on the type of engine, low superheat, up to 50 degrees is used if it is simply desired to dry the steam; medium superheat, up to 150 degrees may be used in engines with sliding valves, and high superheat up to 250 degrees in turbines and special reciprocating engines. Impulse turbines with their ample blade clearances allow somewhat higher superheats than do reaction turbines.

The 250 degrees of superheat means an actual temperature of nearly 650 degrees. This is the highest superheat thus far attempted, for referring to Fig. 3, which shows the effect of high temperatures on the tensile strength of certain metals, it will be seen that a rapid reduction in the strength of steel and cast iron would result if the temperature were carried any higher. Fig. 4 shows the effect of high temperatures on the resisting force between lap-welded steel pipes and their flanges of cast and open-hearth steel.

TYPES OF SUPERHEATERS

The separately or directly fired superheater, one superheater for many boilers, is used but little today; when fired with coal due to its large radiation loss (bad heat utilization) and the difficulty of operating on fluctuating loads. This difficulty is due to the large amount of heat stored in the brick walls surrounding the superheater, which continue to give up the same amount of heat, to the reduced quantity of steam in the superheater, regardless of any reduction in the heat of the flames. The temperature of the steam may thus rise beyond control and completely burn out the lubricatior of the engine or fatally distort the turbine casing. This type, however, is still used in steel mills where waste gases are to be utilized, or where oil firing is used.

The boiler-draft superheater, placed in the current of hot gases in the boiler, is cheaper in first cost and in operation, and adapts itself more automatically to the fluctuations in the power demand, so that this is the modern arrangement of the superheater and the only one here to be considered.

SUPERHEATER CONSTRUCTION

The Schwoerer type of superheater with cast-iron pipes 7½ inches inside diameter, provided with cross ribs outside and longitudinal ribs inside, is simple and reliable, being but little affected by uneven heating (by warming up and by sudden shutting off of steam). The greater area of the outer ribs is required because the transfer of heat from the hot gases to the wall is more difficult than from the wall to the steam.

The seamless-tube type, with tubes usually 1 inch to 1 1/16 inches inside diameter and 0.16 inch to 0.20 inch thick, is more common of late and is cheaper than the cast-iron type; but it must be started up with care. The circular form, wound in a plain spiral or in a helical coil, brings the steam particles to whirling, so that the moisture is thrown against

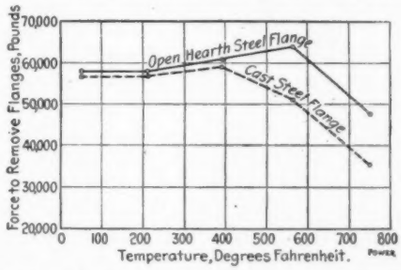


FIG. 4. FORCE REQUIRED TO REMOVE FLANGES PRESSED ON HYDRAULICALLY FROM LAP-WELDED STEEL PIPE

the heating wall. Change of direction improves the transfer of heat from the wall to the steam. Straight and U-formed tubes are also used. If vertical, they are difficult to drain of the water, if horizontal, volatile ash is apt to collect on them. In all cases, they must be perfectly free to expand.

Twisted, cross-shaped bars rolled into the tubes reduce by one-half the maximum distance of a steam particle from the metal wall and increase the heat transmission by 40 to 50 per cent.

COUNTER OR PARALLEL FLOW

As long as the moisture carried over with the saturated steam (about 2 per cent. for a fire-tube boiler and from 3 to

4 per cent. for a water-tube boiler) exists, it is best to use a countercurrent action. It tends to shorten the life (durability) of the superheater, especially with flue-gas temperatures above 900 degrees Fahrenheit, but reduces the heating surface required by 15 to 30 per cent.

To insure effective action of the heating surface, dead corners of the hot-gas currents should be avoided. For superheaters subjected to heavy duty, complete elimination is recommended.

Attention Required by a Superheater

Firing up causes great stress in a superheater, because it is heated by the hot gases without being cooled by any flowing steam. If the superheater lies in a compartment, the hot-gas currents should be directed away from the compartment by closing a swinging or sliding damper of fireclay or cast iron. Steel superheaters not lying in compartments can be protected against burning by flooding with water. This should be carefully drained off so that no water is carried over into the engine or turbine.

Cleaning is desirable once or twice daily as soon as the temperature of the superheated steam begins to drop; this may be done by powerful jet blowers which remove from the tubes the soot and volatile ash which affect the heat absorption to a high degree. The cleaning holes in the brickwork must close tightly. If the cleaning of the soot and volatile ash from the main boiler tubes is neglected, the hot gases reaching the superheater will not be cooled as much as they should and may overheat the superheater tubes, thus causing them to split. Even with the maximum forcing of the boiler, the hot gases on reaching the superheater should not exceed 1500 degrees.

The pyrometer in the flue-gas draft and the thermometers in the currents of steam should be regularly observed and the safety valve should be tested for proper operation. If regulation of the steam temperature is required, it may be done either by adjusting the damper shutting off the superheater compartment or by mixing the steam from the superheater with the saturated steam from the boiler. For the latter, powerful stirring is needed for both kinds of steam. The method of injecting purified water into the superheater is now but seldom used.

Amount of Heating Surface

The proportioning of the heating surface of the superheater is often done from data obtained from practice, especially in case we deal with a plant similar to several previously built. In such cases it suffices to estimate the superheater heating surface on the basis of the boiler heating surface and the grate area. Fig. 5 shows the required superheating surface in percentage of the boiler heating surface of a normally operated water-tube boiler for any desired degree of superheat. It is based on the assumption that one-half of the boiler-heating surface lies in front of the superheater. Here the percentages range from 10 to 40, while some authorities recommend from 10 to 25, if less boiler surface is previously used.

For internally fired boilers, a superheater heating surface of from 50 to 100 per cent. of the boiler heating surface is required. For long-flaming, bituminous coals with much volatile matter, the heating surface of the superheater can be chosen somewhat smaller.

In respect to the grate area, the superheater heating surface for internally fired boilers is about 30 times and on water-tube boilers about 10 times the grate area.

The heating surface of the superheater will be smaller, the higher the temperature of the hot gases surrounding it; that is, the less boiler heating surface utilized before the superheater is reached. The superheater, however, should not be placed in the region of the hottest gases, even if the superheater must be made larger thereby, for it is then better protected and lasts longer. Enough boiler-heating surface should be previously utilized, so that the hot gases on reaching the superheater, even by the maximum forcing of the boiler, do not exceed a temperature of 1500 degrees.

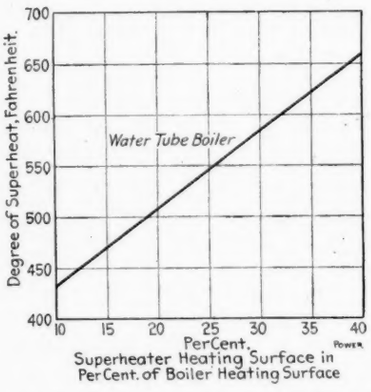


Fig. 5. Relation between Superheating Surface and Boiler-heating Surface

Fig. 6 gives the probable range of the temperature of the hot gases on entering the superheater for normally loaded water-tube boilers using anthracite, and for any percentage of boiler heating surface previously used. For internally fired boilers of the usual length it can be assumed that the entering temperature of the hot gases is from 900 to 1100 degrees. For heavy forcing and for bituminous coal, the temperature of the hot gas on reaching the superheater will be higher.

Where the data derived from practice do not suffice, it is possible to figure the surface from the formula

$$A = \frac{Q}{K \times tm}$$

where

A = Required area in square feet.

Q = Total quantity of heat in B.t.u. to be added to the steam per hour, determined by the aid of Fig. 2.

tm = Difference in degrees Fahrenheit between the mean hot-gas temperature and the mean steam temperature; that is, the mean between the temperatures of the saturated and the superheated steam. The preliminary determination of the mean temperature of the hot gases surrounding the superheater is somewhat uncertain. This will be discussed later in an example.

K = Coefficient of heat transmission; that is, the B.t.u. passing through 1 square foot of surface in an hour for each degree difference in temperature. Its value is somewhat uncertain for the different uses of the superheater, since the influence of the various factors entering into its determination are not as yet sufficiently explained. It appears to depend but little on the velocity of the steam if this exceeds, as it usually does, 30 to 50 feet per second, but probably it increases with the temperature of the hot gases and therefore with the load on the boiler. Powerful whirling of the gases caused by a staggered arrangement of the heating tubes increases the heat transmission. Furthermore, the heat transmission in actual operation is different from that of tests, since the superheater in operation is always covered more or less with soot and volatile ash.

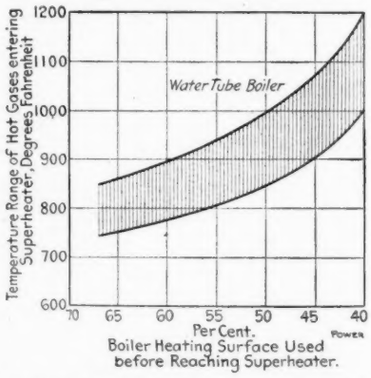


Fig. 6. Probable Temperature Range of Gases Entering Superheater after Passing over a Given Per Cent. of Boiler-heating Surface

K = 2 to 3 for cast-iron superheaters for internally fired boilers.

K = 1 to 5 for steel superheaters for water-tube boilers, depending on the manner of building into the boiler and the intensifying of the boiler-heating surface.

K = 8 to 10 for locomotive and separately fired countercurrent superheaters.

Superheater for Water-tube Boiler

For the purpose of illustrating the formula let it be required to determine the

superheating surface for a given water-tube boiler with the following data:

Steam pressure, 180 pounds gage; boiler heating surface, 2500 square feet; required temperature of the steam, 650 degrees; steam generated per square foot of boiler heating surface per hour, 5 pounds; quality of the steam, 97 per cent.; heat value of the coal, 13,500 B.t.u. per pound; temperature of the hot gases on entering the superheater, 1100 degrees.

The heat to be added per pound of the steam for the superheat (see Fig. 2) equals 147 B.t.u. Since there are

$5 \times 2500 = 12{,}500$ *pounds of steam* generated per hour,

$12{,}500 \times 147 = 1{,}837{,}500$ *B.t.u.* will be required for superheating. To dry out the 3 per cent. of moisture carried over by the saturated steam (whose latent heat equals about 851 B.t.u.),

$0.03 \times 12{,}500 \times 851 = 319{,}125$ *B.t.u. per hour*

will be required in addition, so that the total quantity of heat to be given up to the superheater equals 2,156,625 B.t.u. per hour. Increasing this by 8 per cent. to cover the radiation lost from the brickwork near the superheater, we have the heat to be abstracted from the hot gases

$1.08 \times 2{,}156{,}625 = 2{,}329{,}155$ *B.t.u. per hour*

from which we can now figure the temperature of the hot gases on leaving the superheater as follows:

The coal consumption per hour (on the basis of 8 pounds of steam per pound of coal) equals

$$\frac{12{,}500}{8} = 1562 \text{ pounds per hour}$$

The weight (referred to standard conditions) of the combustion gases developed per pound of anthracite, when using approximately 70 per cent. excess of air, will be 21 pounds with a specific heat of 0.24. The amount of excess air was determined on a similar boiler without superheat from the CO_2 recorder which read 11 per cent. of CO_2 in the flue gas. The flue-gas temperature, t_2, on leaving the superheater is figured from the equation

$$0.24 \times (1100 - t_2) \times 1562 \times 21 = 2{,}329{,}155 \text{ B.t.u.}$$

from which

$$t_2 = 804 \text{ degrees}$$

The average temperature surrounding the superheater equals

$$\frac{1100 + 804}{2} = 952 \text{ degrees Fahrenheit}$$

The average temperature of the steam in the superheater equals

$$\frac{373 + 650}{2} = 512 \text{ degrees Fahrenheit}$$

The mean difference in temperature between the hot gases and the steam equals

$$952 - 512 = 440 \text{ degrees}$$

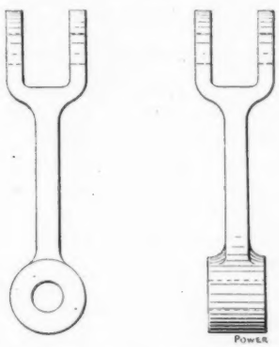
Assuming a coefficient of heat transmission of $K = 5$, the heating surface of the superheater will be

$$\frac{2{,}329{,}155}{5 \times 440} = 1059 \text{ square feet}$$

Wrought Iron Castings

To meet conditions requiring lightness combined with strength, wrought-iron castings are now being used to a large extent. As the name implies, they are made by melting wrought-iron scrap and then casting. The process has been employed in Germany for the past six or eight years, but has only lately been introduced here.

Recently a patternmaker's mistake afforded an opportunity for subjecting one of these castings to a severe test



ROD BEFORE AND AFTER TURNING

In assembling and finishing the pattern for a connecting rod, the bores of the jaws and the eye for the crank pin were placed at right angles. This error was not discovered until after the rod was finished and babbitted ready for assembling.

To save time and labor and without any suggestion from the foundry which made the casting, the rod was heated in a blacksmith's fire and turned 90 degrees to bring the bores in line. There was not the least sign of fracture and the rod is apparently as strong as if it had not been twisted.

Pipe fittings made by this process recently withstood severe tests successfully.

It is claimed that these castings possess all the properties of wrought iron; that they are soft to machine; can be bent hot or cold, hammered, welded and hardened; and show a tensile strength of 50,000 to 75,000 pounds per square inch, with an elongation of 15 per cent. and a reduction in area of 35 per cent. They are made by the Wrought Iron Casting Company, of New York City.

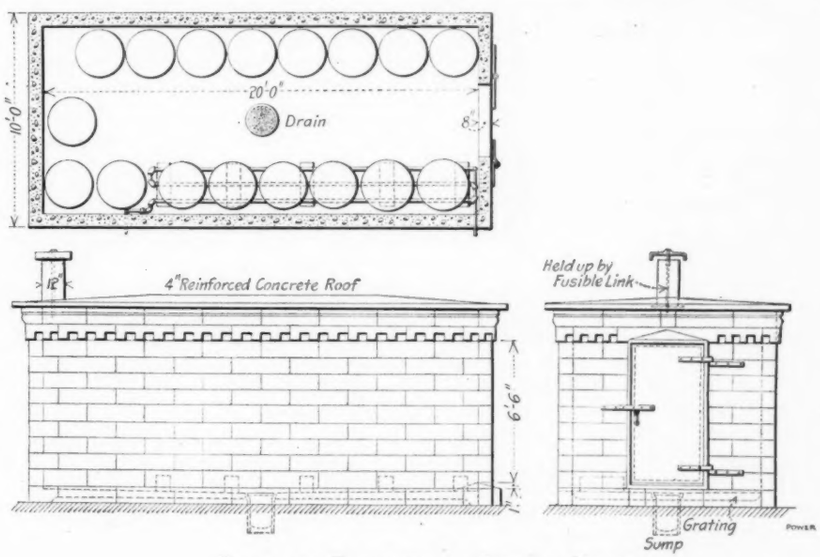
Fireproof Oil Storage House

BY F. B. HAYS

The careless storing of oils, paints and similar inflammable materials increases the fire-insurance rates and exposes the surrounding buildings to a constant danger of destruction from fire or explosion. Fires of this nature are not affected by water, and must necessarily be permitted to burn out. In so doing, they frequently not only burn the materials themselves but also surrounding property, thus causing considerable financial loss.

A Western manufacturing company has designed and built on the grounds of its plant, an oil house for storing such inflammable materials, which it claims will not only prevent the spread of the fire, but will also put it out.

This structure is built entirely of concrete, except for an air-tight steel door and a steel ventilator on top, as may be seen by reference to the accompanying illustrations. The door is so hung that its weight keeps it closed, and the only possible ingress or egress of air is through the ventilator on the roof. This ventilator is held open by a piece of fuse wire, and in the event of fire starting in the house, the fuse wire would melt and the ventilator would automatically close, preventing the escape of the gases and smoke and the admission of air necessary for further combustion, which, of course, would cause the fire to soon die out.



PLAN AND ELEVATION OF STORAGE HOUSE

Power from Compressed Air

By H. Macintire

Air transmission in pipe lines. Developing power from an air system. The economy obtained and the possible applications.

In the transmission of compressed air local conditions are the all-important factor, and these conditions will affect the laying out of the pipe line to the same extent as in power-station design in different parts of the country.

The loss of head or pressure has been found to be proportional directly to the density and the length of pipe, as the square of the volume discharged and inversely as the diameter in inches. In other words, the economy of transmission depends, exactly as in the transmission of direct-current electricity, on how much capital is to be tied up in the first cost. For example, in driving the Jeddo mining tunnel a 6-inch main was used to convey air power to two 3¼-inch machine drills over a distance of 10,900 feet and the loss of pressure was only 0.002 pound, a practically negligible loss. However, it would not be economy usually to design a pipe for such low velocity of the air, as the interest and depreciation on the additional investment over the cost of a smaller pipe line would more than counterbalance the saving in fuel, unless a future demand should make a decided change in the conditions.

In designing the transmission line, therefore, reasonably definite consideration must be given to the future. The pipes, as a rule, are run underground, and are difficult and costly of access. It costs to pass a certain volume of air through a length of 1-inch pipe over three times the head necessary to carry the same volume through the same length of 2-inch pipe, for the periphery increases as the first power and the area as the second power of the diameter. Therefore, as the demand comes on for extra power and an extra pipe is required, the loss of head in the two pipes would be greater than the loss occasioned by a single pipe of an internal area equal to the sum of the areas of the two pipes. The ratio of the periphery to the area of the transmission pipe is the important point affecting friction loss of head. Besides the diameter, the factors affecting loss of head are: The condition of the inner surface, the kind of joint employed, the number of valves and bends, and other factors of like nature. Although a number of tests on the mains in Paris and elsewhere have been made, the data obtained have not been full enough to enable any but approximate calculations. The allowable velocity, however, was clearly brought out. In each case with an initial pressure of 100 pounds, it was found that a loss of 2.4 pounds per mile in the pressure occurred with a velocity of 25 feet per second, 9.4 pounds per mile with 50 feet per second, and 46.2 pounds per mile with a velocity of 100 feet per second.

Many of the precautions taken in laying out a steam-pipe line are required for air transmission. The joints must be carefully made so as to prevent air leaks and to eliminate friction as far as is possible; allowance must be made for expansion and contraction, especially if the pipe is carried above ground; pockets in the line without means of emptying the segregated moisture must be avoided, and, finally, provision must be made for repairs on the pipe should these be necessary.

Some time ago, when the question of air versus electric power was being considered, one important argument in favor of air was that the steam engine could be used with but slight changes in the valve gear when operating with air instead of with steam as the working medium, whereas, of course, electric power requires absolutely new machinery. The indicator diagram for air is almost identical with the steam diagram. When used, however, as is done now almost entirely, to drive special tools, this argument in favor of air will not hold, as pneumatic tools have been designed for air power only.

In general, then, it can be said that the air motor, or machine, is one specially designed for the working fluid. The pneumatic tool cannot be easily described because of the great diversity in the varieties of makes. It uses, however, a pump diagram; that is, it takes air for the whole stroke, exhausts at the end of the stroke, and in consequence is not economical.

Moisture in the air has harmful effects during expansion unless some means can be had to prevent the temperature from going below 32 degrees Fahrenheit. During expansion the temperature drops, the expansion being almost exactly adiabatic, to a greater or lesser degree, according to the conditions. With an initial pressure of 75 pounds, and using a pump diagram, the discharge temperature will be — 60 degrees Fahrenheit, but when expanding to the back pressure an economical diagram will be obtained and the temperature will be — 144 degrees Fahrenheit. This reduction of temperature is very inconvenient because of the impossibility in practice of removing all the moisture in the air, and of the remainder freezing during exhaust. This fall in temperature can be prevented by injecting steam into the air at admission or by reheating. In the case of the addition of steam, its latent heat is given up during expansion and the temperature of exhaust can be kept above 32 degrees Fahrenheit. However, in many cases steam is not available; if it is available it can be used to drive the motor itself.

The second method—that of reheating—is very practical. A coil of pipe similar to those used for superheating steam is usually placed over a coke or charcoal fire and the air is increased some 300 to 400 degrees in temperature at constant pressure. As, however, dry air is slow in taking up heat from dry walls, water is sometimes sprayed in. The effect is twofold: First, the troublesome fall below the freezing point is avoided, and, second, a great increase in efficiency is obtained. The increase in work is about six times what could be obtained from a first-class steam engine at a minimum first cost. Prof. J. T. Nicholson in experimenting with a 27-horsepower Corliss engine, with air at 53 pounds, found that 850 cubic feet of free air was required per horsepower-hour, and dry reheating to 287 degrees Fahrenheit reduced this to 640 cubic feet, or a gain of 25 per cent. The same test showed that 1.42 pounds of coke per hour were required for each additional horsepower, a result which will compare very favorably with good steam-engine practice.

Economics of Air Transmission

So far the discussion has been confined to the means of obtaining power from an air system and certain problems arising therefrom, but now the economics of its use and an idea of its possibilities will be considered. The best idea of its economy can be obtained from the plant in Paris, which has been very carefully tested.

These tests have made the Paris plant very economical. The compressor has an efficiency of from 75 to 80 per cent., the transmission line of 95 to 98 per cent., and the motor, of the best design, from 75 to 80 per cent. The poorer designs of motors, or those badly worn or adjusted, will show as low as 10 per cent. The Paris plant therefore shows that good economy can be obtained with air as the motive power.

Not only is the economy very high, but the uses to which air power can be put are almost without number. These include all kinds of mining tools, the pneumatic tools used in ship, bridge and boiler construction; pneumatic engines for mining and power-mill traction work,

subway and tunnel work where compressed air is used to prevent the ingress of water; for refrigeration to a small extent, and as a means of pumping water (as in the Pohlé air lift).

The advantages of using air are many: It is cheap; there is no danger of explosion from air alone; it is reliable; no insulation is required, nor will the transmission line heat its surroundings. In mining or other confined quarters the exhaust can be used for ventilation. Air replaced steam at the Cleveland Stone Company's works with a daily saving of about 49 per cent.

The great difficulty is lack of flexibility and large first cost. To design an economical plant, either the demand for power must be definite and unvarying or the gift of prophecy must be in evidence. Besides this, the size of the pipe line and the engine is very much larger than electric power would require for the same power, and the difficulty in maintaining the transmission line is greater.

Air power, however, has its own particular sphere, in mining and quarrying, and in all probability it will be found there for some time to come.

Cylinder Oil Tested for Actual Service

By H. B. Lange

Although the oil showed up well under physical tests, the load was so variable that the usual quantity was not enough to keep the valves operating smoothly.

The amount needed had to be determined from actual tests.

In many cases oil is bought entirely on the basis of its physical test without due regard to its ultimate merit, which can be determined only by its actual performance in the particular case for which it is to be used. The following example points out that an oil, although showing a good "physical test," could not do the work reasonably expected of it because the service for which it was intended was so out of the average that an actual operating test alone was the only proper way to explain why a large amount of lubrication was necessary, and why figures consistent with good average practice did not hold in this particular case. The cylinder oil in question tested as follows:

Specific gravity	26.10
Flash point	560 degrees Fahrenheit
Viscosity	170 seconds
Cold test	35 degrees Fahrenheit

It was used on a 16x18-inch single-cylinder, noncondensing four-valve engine, running at 220 revolutions per minute, and direct-connected to a 135-kilowatt direct-current generator. The object of the operating test was to determine the least quantity of oil which would properly lubricate the valves and it was a matter of considerable effort to arrive at a fixed quantity which could be recommended for general running conditions.

The unit was one of three supplying light and power to an office building. The tests were run during two successive days when the average lighting load was around 300 amperes, but the elevators when thrown in produced a swinging load which often reached as high as 600 amperes for a few seconds at a time. Such an irregular load made a lubrication problem which was out of the ordinary. Fluctuating loads reaching nearly 100 per cent. for a few seconds at a time caused such a reaction on the governor that the resulting increase in valve travel demanded a greater quantity of oil than when the engine was carrying the normal load. These overloads, while irregular, occurred at intervals ranging from 40 seconds to 2 minutes. Such conditions led to a necessarily large oil consumption which, on first thought, seemed to be very much against both the engine builder and the oil man. It was because the oil dealer saw no reason why more than one pint of oil in 10 hours should be used that the tests were made.

In conducting the two tests which were made on this oil the matter of first importance was not to cut down the oil so low that the valves would be cut or injured in any way. The "feel" of the valve rods and the sound of the valves themselves were the only indications by which it was possible to determine whether or not they were getting all the oil necessary. A distinct vibration or grinding could be felt on the valve rods the moment the valves were running dry, this being caused by their rubbing in the bore. There was no way of telling whether an excessive amount of oil was used; hence the only way was first to cut the feed so low that the valves ground slightly and then to gradually increase it until these indications disappeared. It was noticed at the start that, although just sufficient oil was used to run the valves smoothly for the normal load, the moment an overload occurred, thus increasing the valve travel and making a larger rubbing surface in the bore, the valves would grind at once. Then it was necessary to increase the feed until the valves ran just freely on the overloads.

Whenever there was any grinding it was noticeable that the greatest friction was in the crank-end exhaust valve, with less in the head-end exhaust valve, while the admission valves ran smoothly at all times. For this reason the crank-end exhaust valve was the one which had to be watched the closest.

In order to determine further whether enough oil was used, the head and admission valve and the crank-end exhaust valve were taken out before and after each run and the wearing surfaces of the valves and the bores were closely examined for any marks of excessive wear.

Oil was fed by means of one of the common forms of forced-feed cups, holding one quart, the stroke of the pump being regulated by a thumbscrew and locknut on the pump spindle. The comparative rates of feed were measured by holding a rule against the locknut as it moved up and down and the travel noted. This travel was equal to the stroke of the pump and varied during adjustments from $\frac{1}{16}$ to $\frac{5}{16}$ inch. The total quantity of oil consumed was measured by filling the cup to a certain mark when the tests were started and refilling the cup to the same mark at the end of the tests, using a measure of known capacity.

The first test was largely experimental, various feeds being tried, but a five-hour run was made, during which time the average stroke of the pump was $\frac{3}{16}$ inch and 1.928 pints of oil were used, which was equal to 3.856 pints for a 10-hour run. The valves on examination showed no signs of wear after the test and the oil was well spread over them.

The second run, conducted the following day, lasted 10 hours when the average stroke of the pump was $\frac{1}{16}$ inch, and 1.948 pints of oil were used. During this run the feed was cut down to the lowest limit and at times the crank-end exhaust valve ground slightly. After the test, when the two valves were removed for inspection, the head-end admission valve showed slight signs of wear, but the oil was well spread over its wearing surface. The crank-end exhaust valve could not be removed without using force. This showed that it was too dry and did not have enough oil spread over its surface, and it had to be left to cool overnight. In the morning the valve was taken out and showed two distinct spots of wear that were not there before the run.

From these tests it was found that in the first run 3.856 pints of oil in 10 hours were more than enough, while in the second run it was evident that 1.948 pints were too little. Therefore, the testing engineer recommended that about three pints in 10 hours be used. This was equivalent to a pump stroke of about $\frac{1}{8}$ inch and with a hydrostatic sight-feed lubricator would be about 3½ drops per minute.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment

Catechism of Electricty

Incandescent Lamps

1135. *How is light produced in an incandescent lamp?*

By passing a current of electricity through a conductor so as to heat it white hot. The conductor that is heated is called the "filament," because it is in the form of a thread. The filament is made very small in cross-section in order to make its resistance high and thereby reduce the current required to heat it.

1136. *Does not the filament burn away rapidly from being kept white hot?*

It would if exposed to the air, because substances heated to a burning temperature in air are readily destroyed by oxidation. In order to prevent this, the filament is mounted within a sealed glass bulb from which the air has been pumped out.

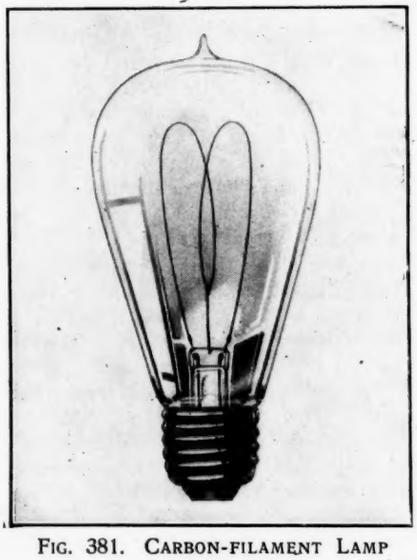


Fig. 381. Carbon-filament Lamp

1137. *What is the usual construction of the lamp?*

Most incandescent lamps are made in the form shown in Fig. 381. The lamp comprises three parts, a glass bulb, the filament and a metal base.

1138. *Of what is the filament made?*

The ordinary carbon filament is made of cellulose—a sticky substance composed of cotton and zinc chloride—which is forced through a small hole to give it the form of a thread. After being hardened, it is cut to the proper length, wound on a form to give it the proper shape, and carbonized.

1139. *What determines the thickness of the filament?*

The size of the hole through which the cellulose is forced. This is proportioned according to the required candlepower, voltage and current of the lamp filament.

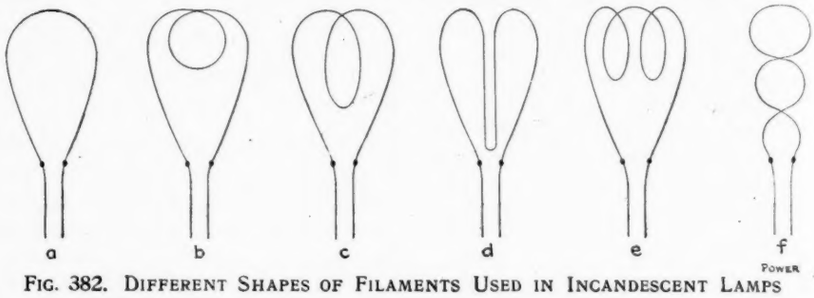


Fig. 382. Different Shapes of Filaments Used in Incandescent Lamps

1140. *In what shapes are filaments made?*

The shape may be either a plain loop as at *a*, Fig. 382, a circular loop as at *b*, an oval loop like *c*, or there may be a combination of two filaments joined in series as shown at *d*, or three filaments in series as at *e*, or again the filament may be made in spiral form as shown at *f*.

1141. *How is the filament supported inside the lamp?*

Two thin platinum wires *c* and *e*, Fig. 383, are fused through a small stem of glass which is solid at *n* but hollow at *m*, and the ends of the filament are fastened to *c* and *e* by carbon paste. Sufficient of this paste is used to prevent the heat of the filament when the lamp is lighted, from harmfully heating the platinum wires. To afford additional security, the filament is anchored at *a* to a small platinum wire fused into the glass *n*. Small copper wires *r* and *s* are fused to the ends of the platinum leading-in wires *c* and *e*, for connection to the base. The glass stem *n m* is fused into the neck end of the lamp bulb and the air is pumped out through a small tubular extension (*c*, Fig. 384) at the opposite end of the bulb, after which the glass around the opening at this latter end is fused together, leaving a small tip projecting outward, as shown at *d*, Fig. 385.

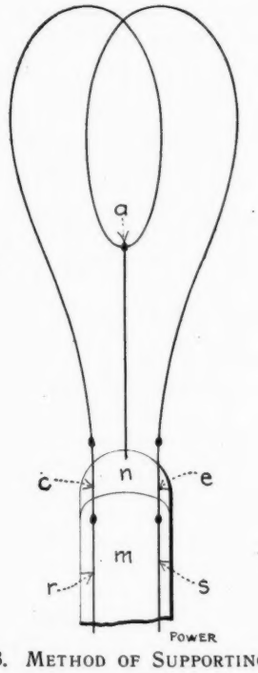


Fig. 383. Method of Supporting Filament

1142. *Why is platinum used for the leading-in wires?*

Because it does not corrode, and because the extent to which it expands under the influence of heat is practically the same as that of glass. There is therefore no danger of the glass becoming cracked or of its expanding away from the wires and allowing air to leak into the bulb.

1143. *How is the base made?*

This consists of a coarse-threaded brass shell *h*, Fig. 386, in the center of which is a metal contact *o* fastened to, but insulated from, the shell by a circular porcelain piece *v*. One of the copper wires leading to the filament is soldered to the brass shell of the base and the other wire is soldered to the center contact *o* so that the two metal parts of the base form the terminals of the lamp.

1144. *How is the base fastened to the bulb?*

With plaster of paris or cement. The rib *i* around the neck of the bulb, Figs. 384 and 385, prevents the glass from pulling out from the base.

1145. *How much light does an incandescent lamp give?*

From ½ to 100 candlepower, and even more, according to the size of the filament. The smallest sizes are used only for special purposes, such as the decoration of rooms and the investigation by physicians of throat and nose conditions. The largest sizes are used for illuminating large rooms, halls, etc.

FIG. 384.

FIG. 385.

1146. *In what sizes are incandescent lamps most generally used?*

The common sizes are 8, 16, 32 and 50 candlepower, the standard size of carbon-filament lamp being 16 candlepower.

1147. *Are there other kinds of incandescent lamps in common use besides those just described?*

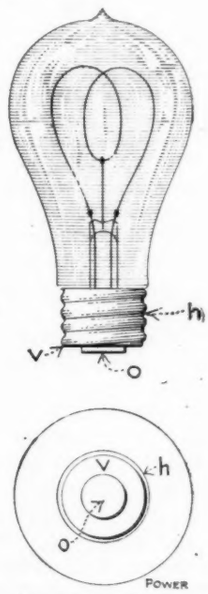


FIG. 386. METHOD OF FASTENING BULB TO BASE

Yes. Besides the lamp with a cellulose or carbon filament, there are metallic-filament lamps of which the tantalum lamp, Fig. 387, and the tungsten lamp, Fig. 388, are in regular use.

1148. *What is the object in making filaments of metal?*

The metal filaments are much more efficient than the carbon filament; that is, they give more candlepower for each watt of electrical power.

1149. *How do the efficiencies compare?*

The high-efficiency carbon filament requires about 3 watts per candlepower of light; the corresponding tantalum filament requires about 2 watts per candlepower, and the tungsten filament about 1¼ watts per candlepower.

1150. *If tantalum and tungsten lamps are so much more efficient than carbon lamps, why are they not used to a greater extent?*

Tantalum lamps and tungsten filaments are not so satisfactory as carbon filaments, especially where there is considerable vibration, because sudden jars and rough handling are liable to dislodge the filament from its mounting and cause the loops to become tangled. Tantalum and tungsten lamps also cost considerably more than carbon lamps of equal candlepower.

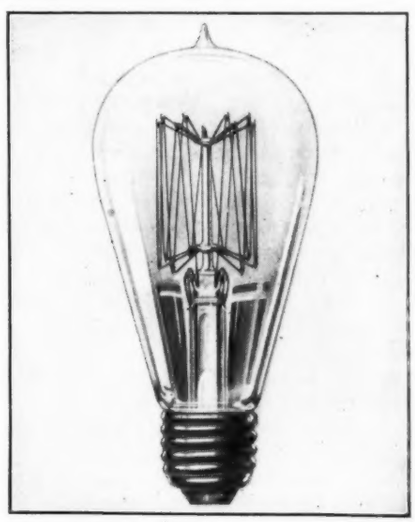


FIG. 387. TANTALUM-FILAMENT LAMP

1151. *Are metal filaments different in shape and mounting from carbon filaments?*

Yes. Owing to the comparatively low specific resistance of tantalum and tungsten filaments, they must be made longer than carbon filaments in order to make the total resistance of the filament high enough. This necessitates mounting the metallic filament upon two spiders each comprising a number of small hooks radiating from a glass support. The filament is loosely wound back and forth over these hooks so as to form loops about 1 inch long in the tantalum lamp (see Fig. 387) and about 2 inches long in the tungsten lamp (Fig. 388).

1152. *How long does the filament of an incandescent lamp last?*

All of the filaments mentioned will last a long time if not subjected to jars or excessive vibration, but as the candlepower decreases with the length of service life, the so called useful life is much shorter than the possible mechanical life. Up to the point where the candlepower becomes reduced 20 per cent. (one-fifth of the original candlepower), the carbon filament has a life of about 400 to 500

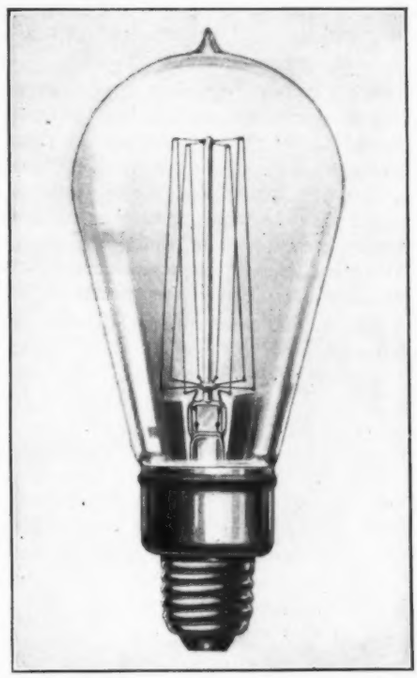


FIG. 388. TUNGSTEN-FILAMENT LAMP

hours, the tantalum filament about 800 hours on direct current or 600 hours on alternating current, and the tungsten filament about 1000 hours on either current.

1153. *As the filaments are in vacuum and cannot burn, why do they give out?*

They are weakened mechanically by the expansion and contraction due to the enormous rise of temperature when lighted and the corresponding drop when the current is cut off. There is also a slow reduction in the thickness of the filament due to minute particles becoming detached from the surface and deposited on the inner surface of the glass bulb by a sort of electrostatic action.

1154. *What determines the choice between tantalum and tungsten lamps, besides the efficiency?*

Tantalum lamps operate best on direct-current circuits. The filaments are better adapted than tungsten to withstand vibrations and shocks because they are not so thin or so fragile. Tungsten lamps operate equally well on direct-current and alternating-current circuits and give a more nearly pure white light than tantalum.

1155. *For what voltages are the usual types of incandescent lamps made?*

Lamps used for ordinary illumination are made for voltages ranging from 27 to 125 volts, but those most used are for 100 to 125 volts because that is the standard range of lighting circuits.

Generating Plants at Victoria Falls

The Victoria Falls & Transvaal Power Company, according to the *Daily Consular and Trade Reports*, has three generating stations in South Africa, the old one at Brakpan and two new ones at Rosherville and Simmer Pan, while the foundations for the station at Vereeniging are now being proceeded with.

The Simmer Pan plant comprises six impulse turbines of 4500 brake horsepower each. The generation is three-phase, 50 cycles at 5000 volts transformed to 10,000, 20,000 and 40,000 volts, as required. The whole system is managed from the control room, which is in charge of an engineer in telephonic communication with every part of the system.

The actual cost of generation is not known, but at the station of the Randfontein Central, where coal costs more in consequence of 25 miles extra haulage, the cost has been brought down to below ½ cent per unit at the switchboard and to 0.8 of one cent into motors all over the property. When the central mill is in full operation and the amount generated increased, with consequent spreading of the standing charges, it is confidently expected that these figures will be improved upon.

The stations of the Victoria Falls Company are being worked to their fullest capacity, the Rand mines requiring considerably more power than they originally contracted to purchase. At some mines, consequent upon the scarcity of hammer boys, the old steam plants have lately been started up to supplement the supply of electricity and compressed air in bulk.

Powerful European Electric Locomotive

A German technical journal reports that the Maschinenfabrik Oerlikon has just turned out an electric locomotive of unusual size and character. It is an alternating-current locomotive of 2000 horsepower, 13,000-kilogram tractive effort, and utilizes current at a pressure of 15,000 volts and a frequency 13 to 17 cycles. The machine is thus larger than that built last year by the Allgemeine Elektricitäts Gesellschaft for the same line; than that built for the Simplon tunnel by Brown, Boveri & Co.; than that built two years ago for the Pennsylvania Railroad, and also than the freight locomotive built for the Italian state railways. It is to be run on the Spiez-Lötschberg-Simplon line. It is from the designs of the engineer, Thormann, and is described by our contemporary as the first demonstration that with single-phase alternating current the highest requirements of existing and projected lines can be satisfied without exceeding the Continental standards with regard to axle pressure.—*Railway and Engineering Review.*

LETTERS

The Right Motor for the Job

I was very much interested in reading the article on selecting the right motor for the job, by Mr. Williston in the October 3 issue, but I do not see why, with the conditions outlined, he did not use the chain drive and motor operating at 1800 revolutions per minute. We have a number of drives of this capacity applied to motors running at 1800 revolutions per minute with ratios as high as 5 to 1 and with the center distances varying between 18 inches and 3 feet. Under the conditions as outlined there is no reason why the drive should not be installed with center distances of approximately 18 inches with a total cost of $96 for the motor and the drive, thereby saving $12 over the cost of the equipment he used and obtaining a higher drive efficiency as well as higher motor efficiency. It is doubtful if a good belt drive can be secured with the 1200-revolution motor to a 500-revolution shaft with 6-foot center distances. We have found that it is inadvisable to use center distances less than 8 feet with ratios of 2 to 1, and 10 feet on ratios of 3 to 1, unless the belts be crossed, as the belt speed and the standard sizes of the pulleys sold with the motors do not allow the best efficiency of drive, requiring altogether too tight belts and, therefore, comparatively short-life belts, besides being hard on both the motor and the shaft bearings.

I do not agree with Mr. Williston that the standard diameter of pulley on the motor as furnished by the manufacturer should be selected. My experience has been that the regular motor pulleys are too small, giving an average belt speed of 1200 to 1500 feet per minute. The best results as regards economy of operation and first cost are secured by using the highest practical belt speed, up to the point where centrifugal force tends to reduce the adhesion between belt and pulley. This in most cases is between 5000 and 6000 feet per minute. By the use of high speeds the size of the belt can be materially reduced, as can also the face width of the pulleys, thereby reducing the first cost and at the same time, owing to the reduction in the required belt tension, reducing the maintenance cost.

In all belt drives it is advisable to make a very careful study of the area of surface available on the smaller pulley, as this is the limiting feature in most cases. If conditions are such that small pulleys are absolutely necessary, the surface contact can be readily increased through the use of special pulleys, such as those with leather faces or cork inserts, but these conditions are usually special and should be avoided if possible. The cork-insert pulley is working out nicely, however, and can be relied upon to transmit a very much greater amount of power per unit of surface. Long center distances are usually necessary, particularly with large ratios, in order to reduce the tension on both tight and loose sides of the belts, and increase the arc of contact on the driving pulley.

HENRY D. JACKSON.

Boston, Mass.

Rope Drive vs. Electricity for Textile Mills

I quite agree with W. H. Booth in his statements regarding textile mills with reference to the relative advantage of electric and the modern rope drive.* So far as we have been able to determine, in a new plant under conditions of operation as originally installed, the rope drive is slightly superior to the electric. There is, however, always to be considered the possibility of expansion, and it is doubtful if, under the conditions of expansion, the rope drive can be so readily adapted to the needs of the enlarged mill as can the electric, with the result that while originally the rope drive was superior to the electric, finally the rope drive would be very much more costly in operation than the electric. Therefore, it is well to figure carefully what the possibilities are of expansion, and in case rope drive is installed, any future expansion, instead of being an addition to the original mill, should be a complete new mill, thereby retaining all of the advantages of the rope drive and none of the disadvantages of attempting to extend the original drive.

With the electric drive, however, no such difficulty exists. The plant can be installed for an economical drive electrically in the first place, and necessary space left for additional units in the power plant, and any additional apparatus installed in the mill can readily be driven by electric motors at good efficiency, no matter where the addition may come, the only requirement being the installation of the motors and the running of the necessary wires.

As Mr. Booth states, it is impossible to make any hard and fast rules as to what to use or not to use, but each particular plant must be worked out on its own basis. It is manifestly unfair both to the engineer and to the owner to make comparisons between plants of totally unlike character.

HENRY D. JACKSON.

Boston, Mass.

*"Electric Drive for Textile Mills," October 3, 1911.

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men

The Rayner Two Stroke Engine

Since the adoption of the Knight sleeve-valve gasolene engine by the Daimler and Panhard companies for automobile equipment, there has been sudden activity on the part of engine builders, designers and would-be designers, with the object of producing a sleeve-valve engine that will be a satisfactory substitute for Knight's without infringing his patents.

One of the latest of these efforts is illustrated by the accompanying engravings, which give largely diagrammatic representations of the construction involved. The engine is really not a sleeve-valve machine in the usual sense of that term. It is provided with two concentric pistons, one of which delivers power to the crank shaft and the other pumps in the charge. Both of them overrun ports and thereby serve as valves in exactly the same way as does the piston of a three-port engine working on the two-stroke cycle. This does not mean, however, that the Rayner engine, here described, is merely the equivalent of an ordinary two-stroke engine.

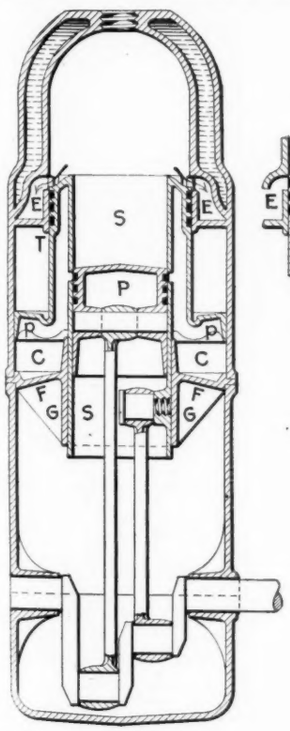


FIG. 1.

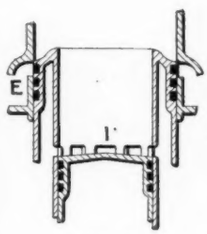


FIG. 3.

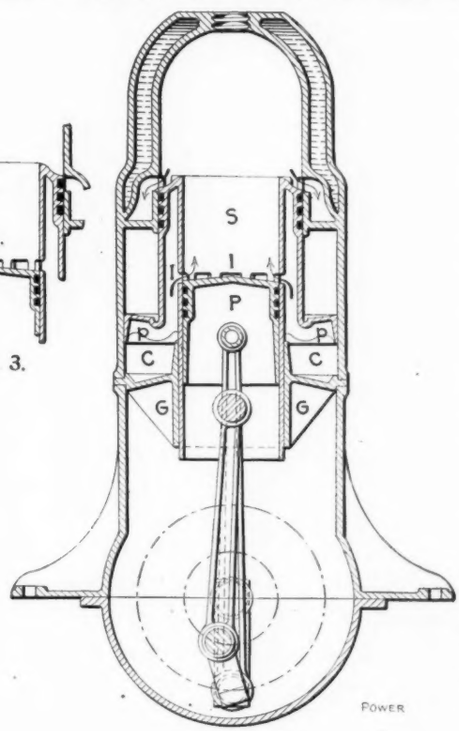


FIG. 2.

Referring to the drawings, the power piston *P* works within the sleeve extension *S* of the pump piston *p*, and is provided with the usual packing rings. The sleeve *S* is attached to the trunk *T* of the pump piston and the trunk slides in the cylinder bore like an ordinary trunk piston and is provided with packing rings near the upper end. The trunk *T* serves as an exhaust valve, covering and uncovering the ports *E*, and the main piston *P* serves as an inlet valve coöperating with the inlet ports *I* in the wall of the sleeve *S*. This sleeve works in a guide *G* which is attached to the outer cylinder wall by a circular flange *F*, thereby forming a closed chamber *C* in which the pump piston *p* operates.

The sleeve *S* and, consequently, the piston *p* are reciprocated by means of an auxiliary crank and connecting rod, as indicated clearly in Fig. 1; the stroke of the sleeve is one-half that of the main piston *P* and the relative motion of their contiguous surfaces therefore occurs at one-half the rate of the piston speed.

Fig. 1 shows the sleeve *S* at the bottom of its travel, with the exhaust ports *E* entirely uncovered. The inlet ports *I*, however, are not yet uncovered because the small crank is set slightly ahead of the main crank, as shown in Fig. 2, and the piston has not yet reached the end of its downward stroke. When it does so, a little later, the inlet ports are uncovered, as shown in Fig. 2, and the fresh charge, which was compressed in the chamber *C* by the piston *p* on its downward stroke, expands into the cylinder, as indicated by the arrows. Its direction being upward, it assists the escape of the burned gases from the interior of the sleeve *S*. On account of the lead of the short crank, the exhaust ports are covered before the inlet ports, as represented in Fig. 3, and there is an appreciable period during which the charge expands into the cylinder after the exhaust ports have been closed.

The continued upward motion of the piston within the sleeve closes the inlet ports and compresses the charge in the cylinder in the usual way. The simultaneous upward travel of the piston *p* draws another charge into the chamber *C* through a check valve (not shown), ready to be compressed on the next downward stroke.

It is evident from the foregoing that the engine operates on the two-stroke cycle, the exhaust and inlet ports being uncovered at the end of each downward stroke. The operation differs from that of the ordinary two-stroke engine, however, in that the inlet ports are open a much longer time, because of the slightly differential travel of the sleeve *S* and the piston *P* during that part of the cycle; for the same reason, the exhaust ports are opened and closed appreciably ahead of the opening and closing of the inlet ports, and they remain open twice as long as in the ordinary engine because the sleeve *S* travels at one-half the speed of the piston.

We understand that the engine has not been built, and therefore no definite information regarding its performance is available. The drawings were supplied by Arthur J. Herschmann, of Watchung, N. J., who represents the inventor in this country.

Chart for Reducing Gas Volumes to Standard Conditions

By J. Albert M. Robinson

In order to do away with the tedious work of reducing gas volumes to standard conditions, the accompanying chart was calculated and plotted. Each diagonal line shows the variation of volume with pressure for a certain constant temperature. The range of the chart is from 32 to 100 degrees and from 27 to 32 inches mercury pressure. It is calculated on the basis of standard conditions of 29.92 inches of mercury and 62 degrees Fahrenheit. The equation is

$$\frac{522}{29.92}\frac{p_m}{T} = V$$

where

p_m = Absolute pressure observed, in inches of mercury;

T = Absolute temperature observed (Fahrenheit temperature + 460);

V = Volume under standard conditions corresponding to one cubic foot under the observed conditions.

Example

In the wet-test meter of a gas calorimeter the temperature is 72 degrees and the pressure is 1 inch water; barometer, 29.69 inches. The low heat value of the gas as metered is 122.1 B.t.u. Required, the low heat value under standard conditions. One inch of water equals 0.07 inch of mercury; therefore the observed pressure is

$$29.69 + 0.07 = 29.76$$

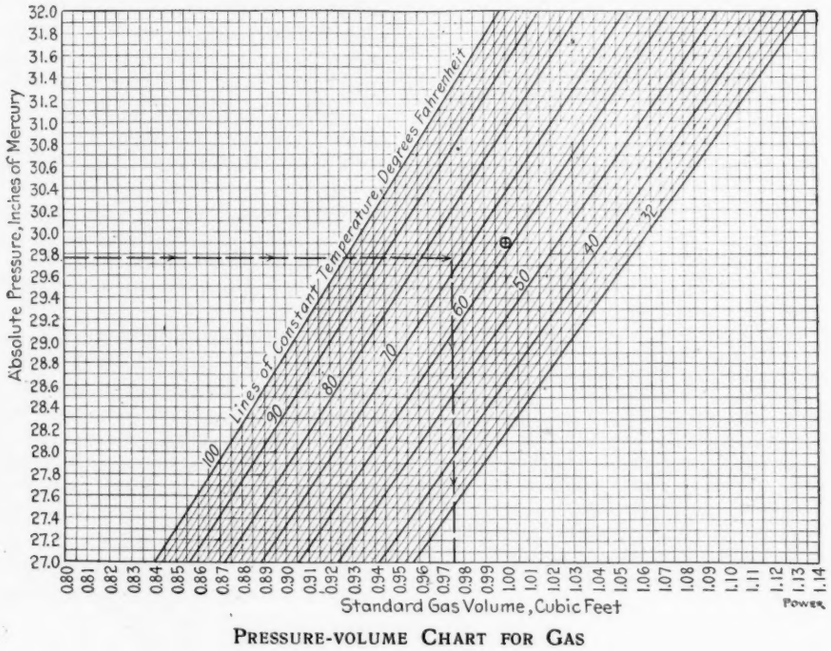
inches of mercury.

From 29.76 on the side scale of the chart, trace horizontally to the right until the diagonal line of 72 degrees is reached; then trace straight downward, as indicated by the dotted line, to the scale of volumes, where the value 0.976 is obtained. This means that one cubic foot as metered is equal to 0.976 cubic foot under standard conditions. Therefore, the low heat value under standard conditions is

$$\frac{122.1}{0.976} = 125.1$$

B.t.u. per cubic foot.

The chart may also be used in the reverse direction. Suppose it is desired to find what pressure would be necessary at a temperature of 32 degrees in order that the volume should be the same as under standard conditions. Trace upward along the vertical line starting at 1 cubic foot until the 32-degree line is reached; then trace the horizontal line at the intersection until the pressure scale is reached; the figure there is 28.2 inches, which is the desired mercury pressure. Conversely, the temperature can be found if the pressure is known.



Pressure-volume Chart for Gas

CORRESPONDENCE

Trouble from Long Exhaust Pipe

Mr. Delbert's account, in the August 22 issue, of the failure of an engine to start through excessive length of the exhaust pipe, recalls an experience I had a few months ago. In my case also the trouble was caused by a straight-run exhaust pipe. It was not practicable to shorten this in the manner indicated by Mr. Delbert, but the difficulty was overcome by substituting a 4-inch pipe for the original 2½-inch pipe.

The speed of the engine was fairly high and the excessive pressure in the long exhaust pipe had no time to disseminate before the next exhaust. Hence, the pressure at the end of the exhaust stroke and at the beginning of the suction stroke was considerable, and instead of fresh gas being sucked in when the inlet valve opened, the waste gases in the cylinder blew back through the inlet and air ports and prevented the admission of a fresh charge. The reduction of pressure at the end of the exhaust stroke, after the larger pipe was fitted, was accomplished as had been expected, and the trouble ceased.

Indicator diagrams taken with a light spring gave the clue to the cause of the trouble; the exhaust lines of these resembled that shown in the illustration, the pressure rising at the end of the exhaust stroke, presumably as the exhaust gases going up the pipe caught up with the previous exhaust charge.

John S. Leese.

Manchester, Eng.

Mr. Caton's Diesel Engine Diagram

The Diesel-engine diagram submitted by Mr. Caton in the issue of October 31 is subject to at least one criticism: the power developed by the engine, according to the diagram, was less than the power delivered by the generator driven by the engine. With 60 pounds mean effective pressure and 164 revolutions per minute, a 16x24 cylinder would develop

$$\frac{60 \times 2 \times 201 \times 82}{33{,}000} = 60$$

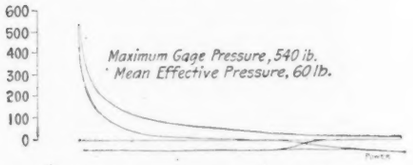


Diagram from Diesel Engine

indicated horsepower. As there were six cylinders, the total indicated horsepower would be 360. If the generator delivered 275 kilowatts, that is, equivalent to

$$1.34 \times 275 = 368\tfrac{1}{2}$$

electrical horsepower, or 8½ horsepower more than that indicated in the engine cylinder. There is evidently something wrong somewhere. The output of the generator should, of course, be less.

Another feature of the diagram that seems to be wrong is the pressure during the exhaust stroke. This was below atmospheric pressure, according to the diagram.

Geo. W. Malcolm.

Brooklyn, N. Y.

"They make engine wheels out of paper now."

"That so? Use 'em for stationery engines, I s'pose."—*Cleveland Leader.*

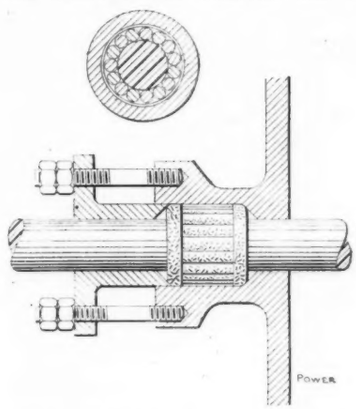
Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words wanted

Repaired Economizer Manifold

A manifold of a fuel economizer cracked just around the fillet to the flange.

The manifold was removed and fastened in a lathe and the inside smoothed up as far as the branch. It was then threaded, and a nipple screwed in place. The nipple can be painted with either red lead or smooth-on.



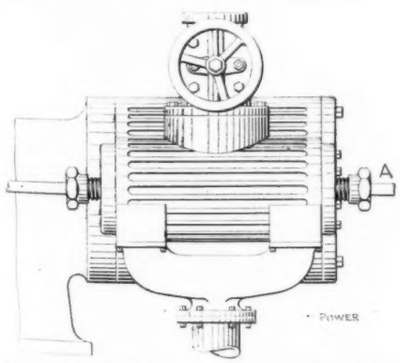
How the Packing is Applied

H. R. Blessing.

Philadelphia, Penna.

Tail Rod Stuffing Box

The accompanying illustration shows the arrangement of a rear-end stuffing box which an engineer placed on a high-speed piston-valve engine. The rear tail rod *A* is held in place by a stuffing box, and holds the valve in a central position; it also reduces the wear on the bottom of the seat.



Stuffing Box for Valve-stem Tail Rod

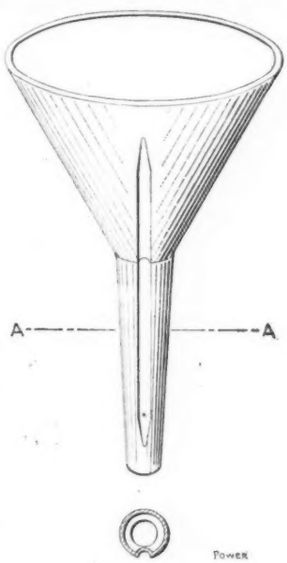
The engineer claims that there has been less leakage past the valve since the change was made and that the valve will last longer because of it.

C. R. McGahey.

Baltimore, Md.

Grooved the Funnel

To prevent tanks, etc., from becoming air bound when filling through a common funnel, I hammered a groove in the side, as shown in the sketch; the groove should be wide and deep enough to prevent its being sealed by the liquid.



Section A-A
Grooved Funnel

P. P. Fenaun.

Lynn, Mass.

Accurate Indicator Connection

Numerous attempts have been made to obtain accurate connections between the engine piston and the indicator driver. Cords cannot serve, nor will chains, as both stretch and a true registry is defeated.

About 45 years ago I discovered a means of absolute accuracy for making this connection and those who desire accuracy can use the old design.

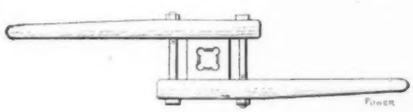
On the end of the crosshead a cut rack was secured to move with the piston. The rack operated on a gear wheel whose axle carried a smaller gear wheel that meshed with a toothed bar which was long enough to operate on one or two geared drums as desired. The rack there gives a perfect registry with every stroke.

Peter Van Brock.

Jefferson, Iowa.

Emergency Die Stock

One way to use a solid pipe or bolt die where no die stock is available is to take two pieces of oak and two ⅜-inch bolts, and use as shown in the illustration.



Wooden Die Stock

J. J. O'Brien.

Buffalo, N. Y.

A Twenty Four Hour Log

The accompanying data are from my daily record. I would like to have the expression of engineers as to the economical operation of the plant.

The plant has a 500-kilowatt turbine which operates with a very small load. The plant, however, has a fair live-steam load. The record for one day's run of 24 hours' duration is given herewith.

LOG SHEET

Average kilowatts	52
Average water temperature at economizer	195 degrees
Average water temperature at boilers	274 degrees
Average water temperature at hotwell	60 degrees
Total water evaporated	382,800 pounds
Total coal used	34,800 pounds
Total ash	3,194 pounds
Evaporation per pound of coal	11 pounds
Average coal per hour	1,450 pounds
Average water per hour	15,950 pounds
Boiler horsepower	462
Cost per 24 hours, including coal, labor and oil	$66.39
Pounds of coal per square foot of grate surface per hour	14½ pounds

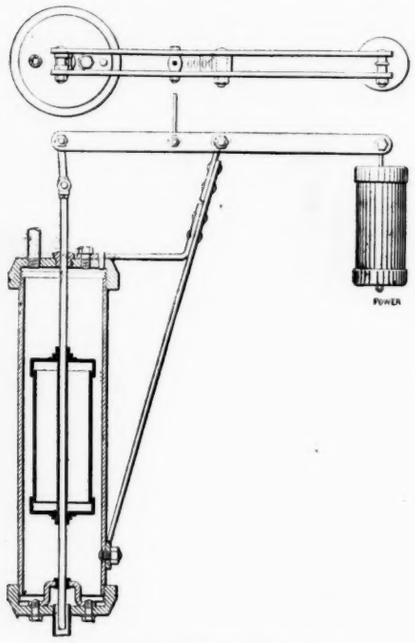
The plant contains two boilers of 275 horsepower each.

C. R. Ward.

Waterford, N. Y.

Float Pump Control

I have had considerable experience with float rigs for governing steam pumps, and have found that the arrangement shown in the inclosed sketch works very well. The float is placed in a separate chamber and connected to the main tank with small pipes after the manner of a water column. The float is made of heavy material and the weight is balanced by a proper counterweight on the other end of the beams. In cases where



DESIGN OF PUMP CONTROL

the weight on the beam center bearing is so great that the float is not sensitive, the trouble is easily remedied by making the beam center bearing ¼ inch larger than the journal and then filling the space all around with pieces of ⅛-inch drill rod, thus making a simple and effective roller bearing.

The beam center bearing was bolted to the bracket in order to facilitate lining; the gear must be accurately in line if it is to work at all.

C. S. RUNION.

New London, Conn.

Removing Piston Rods

There are numerous ways of removing a keyed piston rod from the crosshead. In one plant of which I was chief it was necessary to disconnect the piston rod from the crosshead of one engine several times, the jack shown in Fig. 1 being used to draw the rod out of its seat.

The side pieces were hooked back of the crosshead and the screw which pushed against the rod was turned with a pipe wrench. This worked well but necessitated removing the crosshead pin, which I did not wish to do. While the key was out I took exact dimensions for a set of gibs and key to fit the slot in the crosshead and rod with which to draw the rod without removing the pin. They were made as shown in Fig. 2.

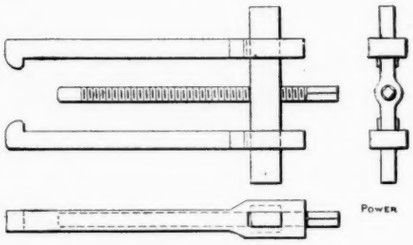


FIG. 1. CROSSHEAD JACK

The key in this case was ⅞ inch thick, 3½ inches wide at the top and 3⅜ inches at the bottom. The wedges and gibs were made so that when they were in place they would be the exact taper of the key in the gib *A*, the shoulders of

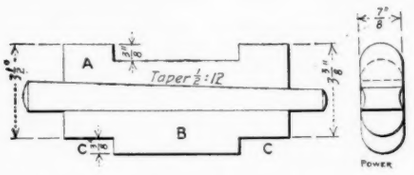


FIG. 2. WEDGE AND GIB ARRANGEMENT

which bore against the crosshead, and the part *B* was cut out ⅜ inch at both ends, as shown. The center part bore against the bottom of the slot in the rod, but the parts *C C* clear the crosshead. The wedge and gibs are given a draw of ½ inch to the foot.

It only requires a few blows with the sledge to loosen the rod. The advantage of this method is that the crosshead pin does not have to be removed when removing the piston rod. The gibs and key are made of machinery steel and were finished upon the shaper.

J. C. HAWKINS.

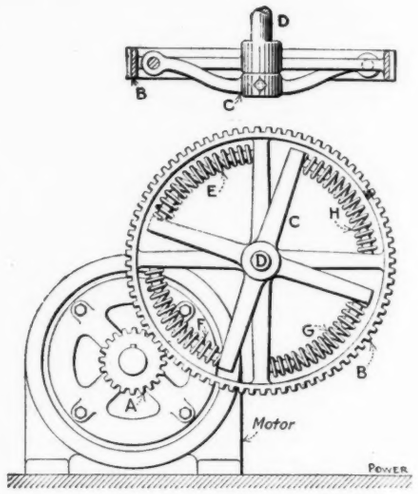
Hyattsville, Md.

Spring Drive

In order to reduce the shock to a machine when suddenly started, a spring drive will be found of benefit.

The accompanying illustration shows a motor with a pinion *A* secured to the armature shaft. This pinion is geared to a large gear *B* which runs loose on the shaft *D*. The spider *C* is secured to the same shaft by setscrews.

As the current is turned on, the gear *B* revolves and transmits its power to the spider *C* by means of springs *E, F, G* and *H*. These springs are held in position by lugs, cast on the arms of both the gear and the spider. The gear starting up suddenly compresses the springs, thereby causing the spider and shaft to start slowly.



A SPRING DRIVE

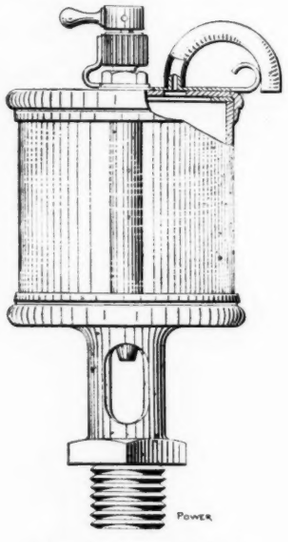
This method of reducing the shock will prevent trouble and increase the efficiency of the machine to which it is applied.

G. B. TANIS.

Paterson, N. J.

Oil Cup Vent Guard

The ordinary glass-sight oil cup has a slide on top to permit filling and the slide has a hole for a vent. To prevent dirt and grit working in through the vent hole, bend a piece of small copper tube having a 1/16-inch hole into U-



AIR VENT IN GUARD

shape and solder it to the slide with the hole in the U-tube over the vent in the slide. This will give the necessary vent and the dirt and grit falling on the oiler can be wiped off—not in.

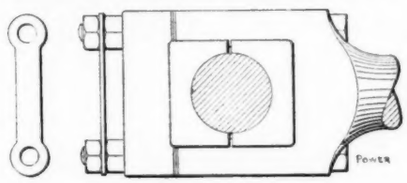
CHARLES H. FRANKLIN.

Schenectady, N. Y.

Adjusting Nut Lock

In cases where the studs in the crank-pin strap are used for tightening the brasses, the scheme illustrated herewith is good.

It is often difficult to tighten the jamb nuts enough without disturbing the ad-



LOCK-NUT KEEPER

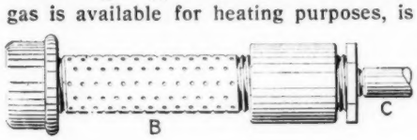
justing nuts. A yoke is made from a piece of sheet iron and placed between the nuts as shown. The lock nuts may then be drawn as tightly as desired without even putting a wrench on the bottom nuts to hold them in place.

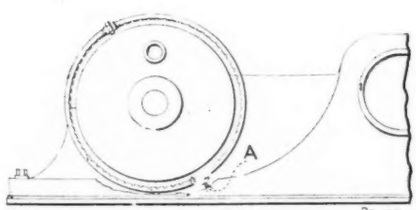
EDWARD T. BINNS.

Philadelphia, Penn.

Replacing Crank Pins

A method employed quite extensively in putting in new crank pins, especially in the larger types of engines and when gas is available for heating purposes, is





HEATING DEVICE

herewith described. The old pin is removed by chipping off the bead on the back end of the pin and driving out with a heavy sledge. After reboring the pin-hole with a portable boring bar, a new crank pin of proper size is made. The usual allowance in present practice for a shrink or press fit for crank pins of from 8 to 15 inches in diameter is from 0.007 to 0.012 inch, and it would require approximately 45 to 125 tons pressure to force them in place if they were put in with a hydraulic press as is the usual practice in most shops.

As in most cases it would be too expensive to rig a portable press even if one were at hand to put in a new crank in the field, most new pins are put in by heating the disk to expand the hole enough to allow the new pin to be put in place and the disk to shrink on the pin.

A method used quite extensively and successfully for heating the disk is shown in the cut. A ring is made of ½-inch iron pipe and bent to form a ring 6 inches larger than the diameter of the crank disk. Small ⅛-inch holes are milled around the inside diameter of the ring to allow the gas flames to be directed onto the disk. A gas pipe or hose is connected at *A*. The gas burner shown at *B* is made from a short piece of 1½- or 2-inch pipe, a pipe coupling cap and bushing and is drilled full of ⅛-inch holes. It is used by inserting it into the crank-pin hole, a gas connection being made at *C*. It is best to let the gas ring outside of the disk burn two or three hours before applying the burner to the crank-pin hole. This allows the disk and counterbalance to become heated up all around, and eliminates the danger of cracking the disk, which would be liable to occur if heat were applied locally to the crank-pin hole.

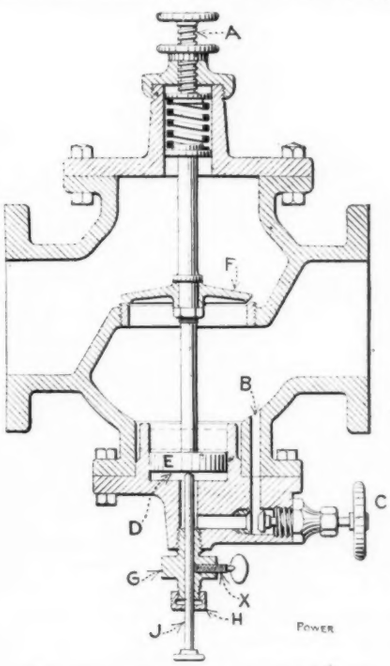
When it is necessary to put on a new crank disk in the field it is good practice to use the crank-pin hole in picking up the disk, and when the disk is in place put in the crank pin, the same heat answering for both operations.

E. P. BAUM.

Pittsburg, Penn.

Troublesome Back Pressure Valve

Several years ago I experienced considerable trouble with a back-pressure valve that would pound on its seat every



IMPROVED BACK-PRESSURE VALVE

time it was opened to the atmosphere, although it was fitted with a dashpot and bypass. It was so designed that when the stem *A* was screwed in the valve and the dashpot piston would rise, it would allow the condensed steam to flow down through the bypass *B*, past the valve *C* and up into the chamber *D*, thereby forming a cushion for the piston *E* and thus preventing the valve *F* from pounding. Although this arrangement looked all right, it did not work satisfactorily.

I proceeded to alter the valve in the following manner: The drip valve *G* was removed and a hole drilled through it and threaded at the lower end for a packing nut *H*. I then drilled and tapped one side to receive the setscrew *X*.

I then replaced the drip valve in its former position, and passed the stem *J* up through it until it touched the bottom of the piston *E*, as indicated by the accompanying sketch. The setscrew *X* was screwed in the side of the drip valve as shown.

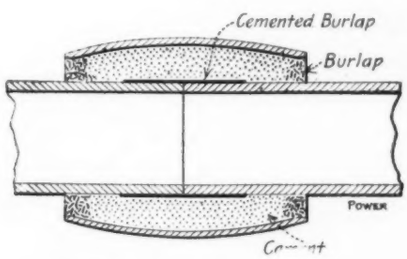
All that was necessary to do now was to slack off on the stem *A*, push up the stem *J* with one hand, and at the same time tighten up on the setscrew *X*, with the other hand. This kept the valve from off its seat and there was no further trouble from pounding.

GEORGE J. LITTLE.

Passaic, N. J.

Concrete Pipe Joint

When about to make some repairs to a headgate it was found that the job could not be done without shutting down the plant, on account of no condensing water. I decided to build a small dam



PIPE JOINT MADE OF CONCRETE

upstream and lay a 6-inch pipe down to the condenser intake. I had 45 feet of pipe, but there was one long piece without threads on one end and no way at hand to cut them.

A nail keg with the heads knocked out was slipped over the pipe after cutting a 2x3-inch hole in one side of the keg. A second pipe was butted against the first one and a piece of burlap 6 inches wide that had been soaked in cement was wound twice around the joint. The keg was then placed over the joint, the ends filled with burlap and cement mortar poured into the hole until the keg was full.

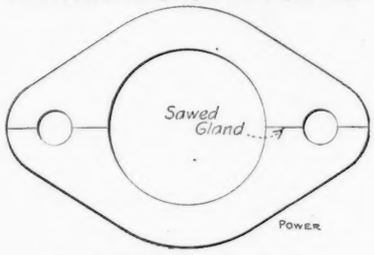
ALMON SKINNER.

Chadwicks, N. Y.

Split the Stuffing Box Gland

Some time ago I was an engineer in a plant where a breakdown occurred on one of the low-pressure pistons of a compound engine.

Before making the necessary repairs I was ordered to prepare the packing. I



SPLIT STUFFING-BOX GLAND

found that the men had forgotten to put the gland of the low-pressure cylinder on the piston rod when assembling the engine. I suggested sawing the gland through the bolt holes, as shown in the sketch.

Two small brass liners were placed between the halves of the gland in the cut. The engine was soon running and the gland gave no trouble; in fact, no changes were made for over two years.

T. D. PARFETT.

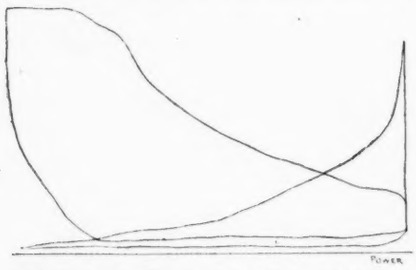
Claridge, Penn.

Governor Gave Faulty Regulation

According to the old settlers on the job, the diagrams shown herewith were the best that could be obtained from a 14 and 26 by 30-inch Corliss engine of the nonreleasing-gear type. It ran at a speed of 150 revolutions per minute and was governed by an automatic governor.

They said that the builder's experts had given up trying to improve the running qualities of the engine which had been hammering away for years.

The governor was pounding at an alarming rate and it was found that the



DIAGRAMS SHOWING FAULTY VALVE SETTING

oil gag pot was inoperative; the piston in it was perforated with four ¼-inch holes.

These free oil passages permitted a too free movement of the governor which caused the gag-pot piston rod to wear.

The four holes in the piston were tapped and plugged, with the exception that a quarter section was cut out of one of the plugs. A new rod was also provided. These were the essential alterations, but they made the governor stiff and yet sufficiently sensitive to give close regulation.

When a governor requires so much resistance against vibration, the logical conclusion is that it is out of balance. Although it may be nearly balanced in respect to centrifugal force of the entire mass of the wheel, it is out of balance in relation to gravitation acting upon the articulate parts.

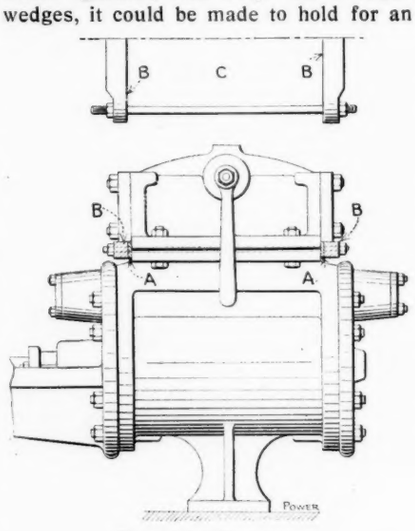
The phenomenon presents a weight rising against gravitation, and with a spring in which a force is stored until the weight descends, when the force stored in the spring acts with the gravity of the weight, thus striking a blow.

V. J. IRONSIDE.

Boston, Mass.

Prevented Packing Blowing Out

A simplex pump, after several years' service underground, had become warped in the planed joints, and on being overhauled in the shops the joint between the steam chest and the cylinder was overlooked. When steam was turned on, the gasket would blow out at the ends. By using different combinations of double gaskets, lead strips and wooden wedges, it could be made to hold for an



PUMP AS REPAIRED

hour or two, but not long enough to keep the water out of the mine.

Existing conditions made it impossible to get another pump, so I devised the scheme shown in the illustration.

After making the 1¼x⅝-inch iron cross bars *B* of suitable lengths, they were flattened on the ends and drilled for ½-inch side rods to form a clamp *C*. I put in a new gasket and a narrow strip of red rubber doubled, across each end as shown at *A*. Over these rubber strips were placed the bars *B*, which were drawn up as tightly as possible by the side rods.

The pump ran in this manner without trouble for 10 months.

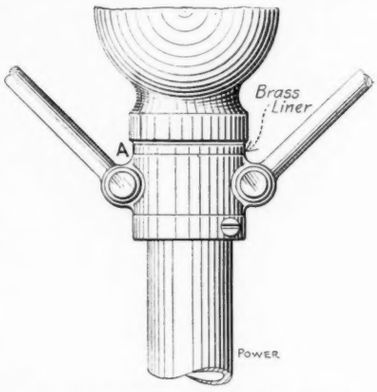
W. E. BERTRAND.

Philadelphia, Penn.

Babbitt in Crank Pin Box

About a year ago I was sent to a mill where the engineer was having trouble with the engine crank pin. I scraped the brasses and polished the pin, but it continued to heat. I then turned the brasses larger and babbitted them, and the trouble was over.

The governor also ran hot, which caused the engine to race and run un-



BRASS WASHER CAUSED TROUBLE

steadily. The governor was fitted with a brass liner, as shown at *A* in the illustration. It was removed and a cast-iron liner substituted and there was no more trouble.

A. L. JOHNSON.

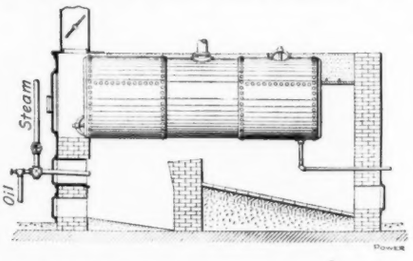
Somers, Mont.

Burning Fuel Oil

Can any reader of POWER give me an approved plan for a horizontal tubular-boiler furnace in which fuel oil is to be burned?

In most of the plants I have seen, the oil is burned with the aid of steam, the atomizer extending horizontally through the front of the boiler setting and in the direction of the draft.

I have been told to let the flame impinge against the bridgewall so that the flame will be deflected toward the ashpit, the grates having been removed; see



FURNACE FOR BURNING FUEL OIL

illustration. Also, to burn the oil with the ashpit door closed and the damper opened a very little.

I would appreciate any information relating to a reader's experience in burning oil, the quantity burned per kilowatt-hour in producing electrical energy.

D. A. STEINER.

Frankfort, Kan.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Engine Runs with Steam Valves Closed

In the September 5 issue, page 371, Mr. Lentz states that he has a 16x32-inch Corliss engine which will run at slow speed when both admission valves are closed, and that after reboring the valve seats, fitting new valves and setting them with the aid of an indicator, the engine runs faster with the admission valves closed than it did before reboring.

He also states that he knows of a 20x36-inch engine of the same type which does the same, although it has only been running six months, and he suggests that the valves are not correctly set; he has tried the latter in all positions with no better result.

Perhaps the following experience will be interesting, as it is a very similar case to his own. The plant consisted of a 20 and 36 by 42-inch engine direct coupled to a dynamo and two small high-speed vertical engines, all running condensing and with 140 pounds boiler pressure. The first extension of this plant consisted of one 300-horsepower and two 1000-horsepower vertical triple-expansion condensing engines. As this necessitated a correspondingly large extension of boiler power, the steam pressure was increased to 180 pounds to obtain better economy. When the Corliss engine was run at this pressure the trouble began.

The parts stood the increased pressure fairly well, with the exception of the crosshead shoe which gave a little trouble through heating. The chief difficulty, however, was that the engine would run above speed when the entire load came off, although the steam-admission valves did not open as the governor was in its highest position.

As the engine had been working some time the valve seats had become worn. They were rebored and new valves were fitted, but with no improvement, for with 180 pounds boiler pressure the engine still ran above speed, although the admission valves did not open.

It was at once realized that this was a question of valve leakage and that the setting of the admission valves would not affect it as they covered the ports correctly when at rest. The admission valves were then fitted most carefully, with but slight improvement.

Then it was decided to run the engine for a time so that the valves could wear tighter. When the vacuum was reduced 4 or 5 inches the engine would govern all right. This was done by cutting down the circulating water, but it needed too much attention as the amount of water used varied with the load and there was also the risk of losing the water, the condenser being of the ejector type.

The boiler pressure was then reduced to 140 pounds whenever this engine was running; at this pressure the governing was quite satisfactory and after a few weeks the engine was tried again at the higher pressure. It still speeded above normal, although the steam-admission valves were not operating; then the makers of the engine were called in.

They sent a man to see what was wrong and in a week or two their expert came. After a week's work he reported that the engine was not designed for 180 pounds steam pressure and must be worked at the lower pressure.

He stated that all condensing Corliss engines could run with their steam-admission valves closed if the engine was first started and the steam pressure was moderately high. He illustrated this by taking out the governor spring, starting up the engine with 140 pounds boiler pressure and showing the engine running with the governor in its topmost position and the admission valves closed.

The report stated that two factors prevented the higher pressure being used in this engine: the high vacuum obtained, 27 to 28 inches, needed very little steam to run the engine light as the condensing plant was not driven from the engine; the other was that the engine was only run at 90 revolutions per minute instead of its designed speed of 115 revolutions. As a consequence, the steam leakage was sufficient to make the engine overrun, whereas had the engine speed been 115 revolutions the governing would have been satisfactory.

Obviously this report left only two alternatives, either valves just large enough for the engine speed of 90 revolutions per minute had to be fitted to reduce the steam leakage or the boiler pressure had to be reduced; the latter was adopted and the engine still runs with a boiler pressure of from 140 to 150 pounds per square inch.

Lest it should be thought that only Corliss engines are capable of being run by the leakage past the admission valves the following experience of a vertical condensing engine fitted with drop valves is given:

The engine was a 20 and 35 by 20-inch fitted with double-beat drop valves actuated by a positive valve gear and designed to run at 230 revolutions per minute; it was sold for direct coupling to a dynamo at 150 revolutions and had an independent condensing plant. At the maker's works it was run noncondensing and the governor acted satisfactorily. When the engine was started up it was found that the governor would not hold the engine when it worked condensing, or when the speed was at about 150 revolutions the steam-admission valves were not lifted from their seats but the speed of the engine gradually increased until the flywheel was endangered.

Many days were spent in the endeavor to make these valves tight. They were ground in cold, and under steam. They were also motor driven and made to hammer on their seats in the hope that they would wear steam tight; then the engine was run noncondensing for a time, but the engine would still speed up, although the admission valves were not lifted from their seats.

It was then decided to put new steam-admission valves in the high-pressure cylinder of just sufficient size to run the engine at 150 revolutions per minute. After this was done the engine was run and the governor then held the engine. Thus it was seen that the difference between the leakage of the larger valves and that of the smaller ones was sufficient to make the engine speed up when there was no load on and when exhausting into a high vacuum.

Mr. Lentz will see from this that his make of engine is not alone in running slowly with the steam-admission valves closed, but it is seldom that there is a combination of high steam pressure, high vacuum, large valves and a light-running engine to demonstrate this fact; therefore it is not commonly known.

JAMES CANNELL.
Stanford le Hope, England.

Show versus Efficiency

An editorial "filler" in the September 26 number, page 490, concerning the engineer who polished his brass work with asphaltum ought to be taken to heart in more than one plant. Too often this showiness in the engine room covers a multitude of sins.

Neatness and cleanliness are virtues to be insisted upon in any plant because clean machinery runs better and because inspection and repairs are facilitated. But cleanliness is just as important downstairs around the filters, condensers and pumps as it is in the engine room.

The engineer naturally likes to receive compliments upon the good looks of his plant, so he proceeds to add polished work to dazzle the eye and draw out more compliments. This is not necessarily a bad thing; but there is always danger that the care of this show work will take time and attention away from the main purpose of the plant, the economical generation of power.

If one looks around in one of these "show places" he will perhaps find a big steam header drained into a well in the rear and all of the water and heat being thrown away. If not this, something equally as bad.

In a State institution some years ago a new power plant was installed. A double system of feed piping for the three boilers was all of polished brass, not even a valve or fitting being painted. Nickeled steam gages were returned and brass ones demanded. Everything possible was of brass, and the fireman was to polish every bit of it every day. Two of the best firemen they ever had left at once on hearing this. One wonders what sort of a CO_2 chart the plant would show with the fireman putting in most of his time rubbing brass. Efficient combustion was evidently a very secondary consideration in this chief's mind, for at all costs things must look pretty.

There were some things that did not look so nice. A 6-inch reducing valve blew out its bonnet gasket and was left in that condition six weeks, leaking steam 24 hours a day. In the engine room was a vertical separator with steam going into the outlet and coming out at the inlet; but how things did shine!

At another plant, one of the largest of its kind in New England, an engineer accidentally spilled a few drops of cylinder oil on the floor. Now the snowy whitness of that floor was an important matter and a grease spot upon it did not blend well with the polished turbines and compressors, so the man was summarily fired. But downstairs the "oiling system" would disgrace a backwoods sawmill. The engines thump and pound, one flywheel imitates the waddle of a duck; but everything shines!

A power-station fireman was one day obliged to paint two exhaust pipes running up the side of the engine room. The next day the chief inquired of him why the coal per kilowatt-hour had risen half a pound above normal. "That," replied the knight of the slice bar, "represents what you saved yesterday by not hiring a painter." He was right. It was impossible for him to fire carefully and at the same time stand on a ladder and paint.

Contrary to the evident belief of many engineers, it is not the fireman who is always puttering at something who is worth the most. Economical burning of fuel demands constant and intelligent attention. Do not distract the fireman; do not think you must keep him on his feet every minute.

The main purpose of a power plant is the transformation of fuel into work in the most efficient manner possible. Then if there is time to primp, go ahead. While the tubes are covered with scale and soot, while the settings are full of cracks, while headers and drip lines and blowoff pipes are leaking and the heater is almost useless for need of a cleaning; while you are buying cold water from the city and pouring hot water into the sewer, no man about the plant can be spared to make a show place of the engine room.

WILLIAM E. DIXON.

Malden, Mass.

Sand for Hot Boxes

Mr. Howarth's surprise and enlightenment, as expressed in the issue of October 17, is only one of the many illustrations of the value of POWER as an educational factor.

Like him, a great many of the younger engineers have spent years at their work without having become acquainted with many of the common practices of older and more experienced men, among which was the use of sand for cooling hot boxes. If any chief on an ocean liner or of any other power plant objects to the intelligent use of sand in an emergency it is because he, like many others, is lacking in knowledge and experience.

I first saw sand used on the S.S. "Roanoke," of the Old Dominion line. The aft journal of the low-pressure engine had given trouble for some time and one day it began to smoke. Water was turned on and the engine slowed down. The chief called for the sand box. When it came the water was shut off, and with one hand he poured a small stream of oil into the box and with the other fed in sand.

The smoke ceased almost at once. In a few minutes the order for full speed was given and the 16-inch journal soon cooled to the normal temperature.

Since then I have often used sand for cases of undue heating in both new and old work, particularly on thrust bearings.

Sand is valuable to the engineer who knows how to use it. It will not cause a journal to heat if oil enough is used to prevent the forming of a paste, but, on the contrary, it will cool a bearing quicker and better than any other remedy that I have ever tried.

Engines are better built and installed now than they were 40 or 50 years ago, and many engineers pass years in service without meeting with some of those incidents which go to make up the sum of a complete practical operating engineer's experience.

W. G. FREER.

New York City.

Engine Knocks

In the October 3 issue, W. A. Mills inquires about a knock apparently in the low-pressure cylinder. If he will come across with a few more data the question could be answered more intelligibly. Does the engine run condensing? How much receiver pressure is carried? Is there any end play to the valves? Is it a Corliss, and what is its size? How much compression is carried and what kind of a diagram is produced?

W. E. CHANDLER.

Northbridge, Mass.

Lifting Water in Boilers

On page 528 of the October 3 issue, C. G. Harden says in substance that, if all the water in a boiler is of practically the same temperature, he cannot account for the circulation, for without temperature differences there can be no circulation. Mr. Harden has a false idea of the cause of boiler circulation and one that is common among engineers.

The sluggish circulation in the hot-water boiler connected to the kitchen range, which is produced by temperature differences, is certainly of a very different character from that produced in a power boiler under full load; the circulation is so strong in the latter case that sheet-steel baffles or other steel parts that may become detached are carried along by the currents produced.

When first starting a boiler, the initial circulation is due to changes in the specific gravity of the contents, due to temperature differences, as Mr. Harden assumes, but as soon as steam is formed the changes in specific gravity become very marked, and from an entirely different cause; namely, the production of steam bubbles, whose weight is very much less than that of a similar volume of water. The tendency is for the water to flow from all points toward the one containing the largest volume of steam bubbles.

All of the water in a boiler under operation is of practically the same temperature unless the construction is such

that pockets are formed away from the paths of the boiler circulation. This is the case in the portion below the flues in a Scotch boiler, unless especially provided with a circulating device.

If it were possible to operate a boiler and maintain the temperature differences of the water at various points, the same as at starting up, the tendency to produce circulation from this cause would be so slight as compared to that produced by the presence of steam bubbles that it could not be considered a factor.

J. E. TERMAN.

Hartford, Conn.

In the October 3 issue, page 528, Mr. Harden has replied to my letter published in the September 12 issue, and I take issue with him on some of his statements. I believe that more explosions are due to improperly cutting in boilers than to low water. But these explosions are not due to the fact that the pressure and temperature rise quickly when the steam pressure is released; they are due to water hammer. Of course, water hammer causes a local rise in pressure at some definite point on the boiler shell, but this is due to the blow given by a rapidly moving body of water and not to any "rise in steam pressure."

Mr. Harden says that my statement that the temperature of the water in boilers is practically constant is new to him. This fact is not new to most engineers. Professor Kent's "Steam Boiler Economy" states that "by increasing the rapidity of circulation in a steam boiler we cannot vary the difference of temperature, for the water and the steam in the boiler are at about the same temperature throughout. The circulation in a boiler is due to a large number of steam bubbles which are mixed with the water. The specific gravity of the mixture is lower in its upward than in its downward movement, and this is what causes the circulation in boilers and not a difference in temperature.

In both of Mr. Harden's communications he has claimed that there is a rise in steam pressure due to opening a valve from a boiler into a header containing a lower pressure than that in the boiler. It is improbable that there can be any rise in steam pressure, as the steam space in the boiler is connected with a header of a low pressure. Therefore the steam would flow from the high-pressure boiler into the low header and prevent any accumulation of steam pressure. Furthermore, where are the heat and energy coming from to "cause the temperature and pressure to rise rapidly"?

Mr. Harden has continually used the expression "flashpoint" of water. This is an exceedingly misused word in this connection, for one would suppose that if water were at the temperature corresponding to the steam pressure, it merely needed a little shove to push the whole mass of water into steam. But this is not the case.

Boiler explosions have been caused by opening the stop valve between the boiler and header *before* the steam pressure in the boiler was equal to that in the steam header. In this case the higher pressure from the steam main is admitted to that in the boiler, but it is not this pressure which causes the boiler to explode, because the pressure in the main may be lower than the normal working pressure of the boiler and still cause an explosion. The trouble is due to water hammer.

Since Mr. Harden wants some information on water hammer, I will add the following: Water hammer has been explained as follows: The steam discharged into the boiler from the main disturbs the surface of the water and generates waves. The moment one of these waves breaks so as to form a sort of bubble and lose some of the steam, the water-hammer action begins, because the water in the boiler is at a lower temperature than that of the steam (due to lower pressure) and it follows that the steam inclosed within the bubble will condense, leaving a space there at a lower pressure than in the steam drum. The higher pressure acting upon the surface of the water will then close the bubble up almost instantaneously, and the water forming the upper wall will be brought down against the bottom with great speed and violence.

The hammering action thus initiated will increase the disturbance in the water and larger and larger bubbles will be formed in the same way, with the production of increasingly violent shocks when they collapse. Soon these waves and shocks attain enough violence to start a rivet and then the pressure within the boiler does the rest. In fact, it is always the normal boiler pressure which does the most damage and it is only necessary for some minor momentary disturbance to start a rivet which the normal pressure will pull apart and then is had the entire range of pressure from that of the boiler down to atmosphere with a corresponding drop in temperature available for throwing the boiler pieces around.

Some idea of this water-hammer action derived from blowing steam into a body of colder water may be obtained by sticking a hose into a barrel of water, and the chattering and pounding which water-hammer action will set up from a ¼-inch opening will give some idea of what would happen when a 6-inch main is discharging steam, although, of course, in the latter case the conditions are not so ideal as the steam would not be discharged below the surface of the water until it got the water surging about violently enough to make its own steam bubbles.

FRANK J. MCMAHON.

New York City.

Steam Engine Lubrication

I read with interest the article by R. D. Tomlinson on steam-engine lubrication in the September 12 issue, page 396, of POWER. It seems to me that the oil companies should not only label their oil barrels with the flash-and-fire test, but they should also stamp on the head of the barrel the temperature at which the oil will properly atomize and mingle with the steam. Then it would be a very simple matter to choose the cylinder oil best suited to the steam pressure carried.

At one time I had charge of a tandem-compound engine and I had considerable lubrication trouble with it. I used the very best cylinder oil that I could buy but my trouble increased. Then I purchased a cheaper oil and got splendid results in the high-pressure cylinder, but the low-pressure cylinder continued to work badly. I then attached a plain cylinder lubricator to the low-pressure valve and my troubles were over. The boiler pressure was 100 pounds per square inch.

CHARLES FENWICK.

Wapella, Sask, Can.

Mr. Rockwell's Questions

Charles J. Mason's quotation from *Railway and Locomotive Engineering*, published in the October 3 issue, page 528, may be all right for locomotives, but I believe it is entirely wrong as regards stationary practice. While putting cold water into a red-hot boiler may not cause an explosion, it will do other damage, especially in water-tube boilers.

About two months ago a water tender in charge of a large battery of water-tube boilers fell asleep and the water dropped out of sight in the glass in No. 2 boiler. When he awoke he opened up the feed valve and the result was that 14 front headers were cracked on a line at the third row of tubes from the bottom and the tubes were also burned above this point, not below, which shows just where the water was when the feed was turned on.

I have seen several other cases similar to this in water-tube boilers, which leads me to believe that water should not be turned into a boiler with low water until it has been cooled down. Cold water will not cause an explosion but it will cause other serious trouble, which in some cases might not extend beyond leaking seams in a return-tubular boiler, but the sudden contraction may cause the rivets to be partly sheared and to give way in an explosion at some future time. A much safer and saner way is to cool down the boiler before turning on the feed water.

J. C. HAWKINS.

Hyattsville, Md.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Shafting Calculations

If a line shaft is to transmit 100 horsepower, how may the diameter be found?

C. E. S.

For turned-iron line shafting a very commonly used formula is

$$Horsepower = \frac{d^3 R}{125}$$

in which

d = Diameter of the shaft in inches;
R = Number of revolutions per minute.

Assuming 300 revolutions per minute,

$$100 = \frac{d^3 \times 300}{125}$$

$$d^3 = \frac{12{,}500}{300}$$

$$d = 3.44 \text{ inches.}$$

The nearest commercial size is $3\frac{7}{16}$ inches.

Steam Discharged from Pipe to Vessel

What horsepower of steam will be discharged from a ¾-inch pipe with 50 pounds at the delivery end discharging into a vessel in which a pressure of 30 pounds is maintained? The pressures are gage; call them 65 and 45 absolute; the end of the pipe is wide open.

J. O'D.

Dry saturated steam at 65 pounds pressure has a temperature of 298 degrees and contains 1172.9 heat units. At 44.67 pounds the temperature is 274, and the heat content after adiabatic expansion from the above condition is 1143.9 B.t.u., a difference of

$$1172.9 - 1143.9 = 29 \text{ } B.t.u.$$

expended in acquiring velocity. This is equivalent to

$$29 \times 778 = 22{,}562 \text{ } foot\text{-}pounds.$$

The energy of a moving mass is expressed by the formula

$$E = \frac{W v^2}{2 g}$$

in which

E = Energy of the moving mass;
W = Weight of the mass;
g = Force of gravity;
v = Velocity of movement.

By transposition and substitution

$2 g E = v^2$ where W = unity;

$v^2 = 2 \times 32.16 \times 22{,}562$;

$v = \sqrt{1{,}451{,}187.84}$;

$v = 1204$ feet per second.

The area of a ¾-inch pipe is 0.0037 square foot and the volume discharged per second will be

$$1204 \times 0.0037 = 4.45 \text{ } cubic \text{ } feet$$

or 16,020 cubic feet per hour. At 274 degrees the weight of steam is 0.1058 pound per cubic foot.

$$16{,}020 \times 0.1058 = 1694.91 \text{ } pounds \text{ } per \text{ } hour$$

which at 30 pounds per horsepower per hour would be

$$\frac{1694.91}{30} = 56.29 \text{ } horsepower$$

Graphite as a Scale Preventive

Will graphite, used as a scale remover or preventive, have any injurious effects on a boiler?

C. B. S.

Several engineers have reported satisfactory experiences with graphite as a scale remover and preventive. There is nothing in graphite which can be injurious to the boiler itself. But in its use care should be taken to prevent injury from an accumulation of detached scale on the heating surfaces.

Compressor Suction Pipe Covering

Is it necessary to cover the suction pipe leading from a refrigerator to the compressor?

J. B. C.

It is good practice as it tends to prevent the absorption of heat by the contents of the pipe and increasing the volume, thus reducing the capacity of the compressor. Covering also tends to prevent the drip of water which comes from an uninsulated ice-covered pipe.

Ammonia Expansion Valves

Will one ⅜-inch expansion line and cock be sufficient for a brine tank containing eleven 1¼-inch double coils 50 feet long?

L. A. O.

It will suffice if less than 100 feet long, though for 1100 feet of 1¼-inch pipe two expansion valves would be better.

Heat Value of Lignites

In burning Dakota lignite I find a great difference in the ash content. It is claimed for one kind making about four times as much ash and clinker as another that its heat value is greater than that which makes but little ash. Can this be true?

L. R. D.

It is quite possible for a coal which is higher in heat units or which will show even less ash upon analysis to make more ash or ashpit refuse in use. An ash which has a low fusing point will melt and run together in the furnace and carry a lot of carbon with it, so that the weight of stuff taken out as refuse is not all ash, and may be greater in quantity with a lower ash coal than it would be with a coal having a higher percentage of real ash which stayed in a powdery form and could be sifted as ash into the ashpit.

Burn a small sample of the coal thoroughly by itself, as in a porcelain dish, completely to ash and find out in that way what the percentage of real ash is and how it compares with the ash in the other. The heat value of the coal may also be higher irrespective of its ash content, depending upon its composition otherwise.

Starting Cross Compound Engine

How may an ordinary cross-compound engine be started when the high-pressure crank is on the center?

H. B. U.

By admitting steam to the low-pressure cylinder. Such engines are usually provided with means for admitting live steam to the receiver. When there is nothing of this kind the throttle valve may be opened slightly and the high-pressure valve gear operated by hand, a few strokes of which will pass enough steam to the low-pressure side to start it. If there is a vacuum in the condenser, the opening of one of the indicator cocks will admit air enough to start the piston.

Blister on Boiler Sheet

What is meant when it is said that a boiler is blistered?

B. B S.

In the manufacture of iron boiler plate it sometimes happens that the layers are not thoroughly welded and the heat of the furnace causes thin, unwelded parts to bulge outward from the plate in the form of a blister, from which the name is derived.

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POWER

Power Plant Records Necessary

In many instances where a change has been made from homemade power and light to central-station service it has been found, when too late, that instead of a reduction in the cost of this service there has been an increase of one hundred per cent. or more.

It is somewhat difficult, if not quite impossible, to apprehend the mental attitude of the business man who, whether acting for himself or as the agent of another, will make changes involving the expenditure of thousands of dollars annually without first engaging the service of disinterested experts in the line in which the expenditure is to be made.

If raw material is purchased for consumption in any line of manufacture, quality and price per unit of measure or weight are among the specifications in the contract, and the buyer gives intelligent consideration to every paragraph in the proposal he is asked to accept, going at times to the length of employing chemical or engineering advice before making his decision.

But when it comes to the purchase of heat, light and power the average business man is all too often ready to accept the unsupported statements of the central-station solicitor without question.

There are badly designed, poorly equipped and incompetently managed isolated plants which cost the owner more to operate than it would to buy the same service from the central station. Such plants, however, are the exception instead of the average, and there are few of the very worst that cannot by the application of power-plant common sense be made to show results in the cost of light and power that the central station will hardly meet.

One of the items on which the solicitor dwells at great length and with much earnestness is the superiority of electric transmission over shaft and belt where power is used for machine driving. Its superiority lies alone in the factor of convenience, and the owner or user who understands will not be deceived by the most eloquent plea along this line. Electric transmission is convenient, but it is not cheap, as many have learned after a change from one system to the other.

When an owner or manager is approached with a proposal to change from one system to another, sound business instinct should dictate that not only should the claims of the new system be as rigidly investigated as would be the commercial standing of a new customer who wished to make a long-time contract, or the merits of a substitute material for manufacturing purposes. Power-plant records, which should be kept as scrupulously accurate as the accounts of any other department, will show whether a change is advisable.

If plant records are not kept the custom should be instituted at once. It is highly probable that the information gained in this way will pay for its cost many times in the improvement which invariably follows a change from a slipshod lack of method to business ways of doing common things.

Boiler Efficiency

When the American Society of Mechanical Engineers' committee on rules for power-plant tests wakes up it might profitably devote a few minutes to Rule XXI of the "Code for Conducting Boiler Tests." This rule states that the efficiency of a boiler is the ratio:

$$\frac{\textit{Heat absorbed per pound of combustible}}{\textit{Calorific value of 1 pound of combustible}}$$

and that the efficiency of the boiler and grate combined is the ratio:

$$\frac{\textit{Heat absorbed per pound of coal}}{\textit{Calorific value of 1 pound of coal}}$$

With no definition of "combustible," these two formulas give precisely the same results. For example, if the coal contains ninety per cent. of combustible, both the numerator and the denominator of the second ratio will be numerically ten per cent. smaller than those of the first ratio, but the results of the two will be exactly equal.

Of course, we are familiar with the methods commonly followed in making boiler tests, and in the light of such familiarity it becomes clear that the "combustible" in the numerator of the first ratio means that part of the fired combustible which is actually burned—that is, does not fall through the grate, but the code does not say so and that is not the definition of the term for any purpose other than boiler testing. Therefore, the first ratio is not stated with abstract clearness. It should read:

$$\frac{\textit{Heat absorbed per pound combustible burned}}{\textit{Heat value of 1 pound combustible}}$$

The second ratio, for overall efficiency, would be improved by adding the word "fired" after the word "coal" in the numerator.

Boilers under Sidewalks

Nowhere is the inconsistency of city, State or Government control over the safety of the public more in evidence than in the matter of boiler installation. On the statute books of every town, city and State are laws prohibiting persons who have no regard for the safety or welfare of others from doing certain things, and such laws are based on sound common sense. Why, then, should not laws be enacted to govern the placing of steam boilers?

Under the present laws of most cities the owner of a steam boiler can put it wherever he pleases, except in the street, and permits are issued even for this purpose. Thousands pass these boilers every day who are ignorant of the danger to which they are exposed from an explosion.

In hundreds of instances the steam boilers of buildings are placed beneath the sidewalk. They are thus installed because they are convenient for coal delivery and because the owner of the building secures a larger basement area. No consideration is given to public safety and few passersby know that the boilers are so placed until the building has been torn down or an explosion occurs.

An example of what may happen at any time, and particularly in a congested area, occurred in New York City recently, when a return-tubular boiler exploded, an account of which explosion has been published.

It is true that for twenty-three years this boiler had been under steam pressure without accident, but one night it exploded and demolished the sidewalk for more than half a block, throwing tubes, bricks and fragments of flagging into the street and the adjoining buildings. It is fortunate that the streets were deserted at the time of the explosion, save for a solitary policeman who was standing on a street corner half a block away, and he was injured.

What would have been the property damage, what would have been the death toll, and how many would have been injured had the boiler exploded at noon instead of at midnight?

Is it safer to put a boiler under the sidewalk than it is to place it in the basement of a building? We do not believe that it is. It is true that the destruction of property might be greater when a boiler in the basement explodes, for the building would be partially wrecked and strained. But when a boiler installed under a sidewalk explodes during the busy hours one might as well train a gatling gun on the people.

It is good fortune that no lives were lost in this explosion, but the lesson to be drawn from it is that greater vigilance should be exercised over all boilers so located, though the chances are that the old lap-seam boilers will continue to be operated under the sidewalk as they have for years past.

Smoke Inspection

Elsewhere in this issue will be found reference to some of the activities of the smoke-inspection department of the city of Chicago. Although this particular instance applies to the marine branch of the bureau, similar steps have also been taken relative to stationary plants.

The policy followed by this department is established on broad and comprehensive lines; instead of arbitrarily enforcing the law and leaving the offender to seek his own remedy, the department cooperates with the latter and gives the benefit of its extended experience.

A similar policy has been adopted by the city of Boston. There the law has established the standards by which the density of the smoke is to be judged; it prescribes what density is permissible, and provides for a definite penalty. By this means the owner or the engineer, by exercising close supervision over the plant, is able to judge whether the law is being transgressed.

Furthermore, the smoke-inspection department has made extensive experiments in smokeless combustion with various fuels and, as in Chicago, coöperates with the plant owners, advising them as to the type of furnace to be employed with different fuels, the methods of firing best adapted, etc.

It would seem that New York City might profitably follow the lead taken by these two cities in the enforcement of such an important ordinance as that relating to smoke nuisances. While it is true that much of the coal in the Chicago markets has more smoke-forming qualities than that available for use in New York, yet the latter does not differ materially from that used in Boston.

In New York there is no distinct smoke-inspection department, the duties of such being performed by the health department. Furthermore, the law merely states that no dense smoke shall be emitted from any stack within the city limits, and provides for fining the offender. The question of density is left entirely to the judgment of the inspector and the owner is offered no assistance in relieving the conditions.

It is now well known that practically smokeless combustion can be obtained with bituminous coal, providing the proper type of furnace and the correct methods of firing are employed. Different coals and boiler equipment require different treatments, however, and the difficulty lies in the fact that the average owner or engineer of the small plant has not had the experience which will enable him to meet individual plant conditions with smokeless combustion when burning bituminous coal. Therefore, he turns to anthracite, which, of course, precludes the use of mechanical stokers.

In view of this, coöperation of the smoke-inspection department (which has the facilities for making a thorough study of all conditions) with the owner is most desirable.

Terminal Pressure and Compression

It sometimes happens that an engine which runs quietly with a certain steam pressure and load will begin to pound if either are changed. This oftenest occurs when the load is increased or the pressure diminished.

As the piston approaches the end of the stroke its velocity decreases rapidly and at the end becomes zero.

While the piston is moving through the cylinder with the steam behind it the pressure on all of the pins and bearings is in one direction and all of the lost motion necessary for lubrication in the reciprocating and stationary joints is taken up in this direction. If near the end of the piston travel the exhaust valve is closed and steam enough is caught in the clearance space to raise the pressure in front of the piston above that behind it, it will tend to reverse the direction of the pressure on the pins gradually and the piston will start on the return stroke quietly. If, however, the compression is not enough higher than the terminal pressure on the opposite side of the piston to take up the inertia of the moving parts, the reversal will be accomplished suddenly by the steam which enters the cylinders as the steam valve opens and the engine will pound unless very closely keyed.

If an engine is running quietly with normal load and steam pressure with a compression sufficiently above the terminal pressure to gently affect the moving parts as they come to rest, and either the load is increased or the steam pressure reduced the terminal pressure in the cylinder will be increased. If the increase in the terminal pressure is high enough to nullify the effect of a slight compression the engine will, unless closely adjusted, change from a quietly running machine to a noisy one.

In the operation of cross-compound condensing engines it often happens that an increase in the receiver pressure will make the low-pressure side pound on the centers because the higher pressure in the cylinder at the end of the stroke brings about a more sudden reversal in the direction of pressure on pins and journal, taking up the lost motion with a thump instead of gradually as is desirable.

It will be found in a great many cases that it is excessive lead rather than the lack of compression that causes the pounding that can only be cured by excessive compression.

Heating and Ventilation

Considered as power plant problems. Layout and operation of systems and apparatus

School Heating *

By Ira N. Evans†

In schools of the high and manual-training class, where rooms aggregate more than 16 and the pupils over 500, engines are included for power purposes with increasing frequency in connection with the use of exhaust steam for heating.

The peculiar conditions of school operation make this arrangement economical, as the lighting and power are required during sessions, and the summer vacation reduces the number of hours to practically the heating season.

The site requirements for light and air reduce the lighting load to periods of school sessions in winter and the fresh warm-air requirements for occupants insure ample boiler capacity for power purposes before utilizing the exhaust in the heating system.

In the article of September 12, the hours of operation for the heating season are given in Table 2 as 5036 for the New York district with 10-hour day periods. Deducting 31 Saturdays of five hours each, or 155 hours, and taking eight-tenths of the result, the number of

Fig. 1A. South High School, Worcester, Mass.

hours for school children during the heating season are obtained. This contemplates an operating-day period of from 8 a.m. until 4 p.m. The hours deducted from day periods should be added to the night and holiday periods and the percentages of Fig. 3 in the September 12 article applied to obtain the number of hours for each 10-degree period of outside temperature. This gives a day operation of 1112 hours and a night and holiday operation of

$$5036 - 1112 = 3924 \textit{ hours}$$

Table 1 gives the number of hours as described. This shows that schools are actually in operation 22 per cent. of the time and outside of public uses, nights, etc., are closed 78 per cent. of the time during the average heating season in the New York district.

The law in most States requires that school buildings be provided with a system of air supply and removal which will furnish a minimum of 30 cubic feet of air per pupil per minute heated to 70 degrees Fahrenheit. In small buildings this has been accomplished with a hot-air furnace and a gravity circulation of air with fair success. When the building in question has over six rooms and

Fig. 1B. Brockton High School

the inspectors insist on a proper air supply, the apparatus becomes so bulky and expensive to operate that this system must be abandoned.

For six- and eight-room schools a gravity steam system is employed with indirect radiators to heat the air supply, which in turn depends on gravity for its operation, with the attendant low velocities and large flues. Auxiliary direct radiation in the rooms may be used as well. Aspirating coils are also used in the ventilating flues to produce a movement of air in moderate weather. This is a wasteful method except on a small scale and at the time most needed—moderate, damp weather—is likely to fail. In cold, clear days the hot air in the room will create sufficient draft without the aspirating system.

In cases where public-service current is available, motors may be used to drive the fan. This reduces the size of flues and gives a positive assurance of results.

In school plants there seems to be a desire to eliminate all apparatus possible that would require knowledge and intelligence in the man in charge, so that the heating plant for large buildings is often designed in the same manner as for small schools. This involves heavy operating expense in the number of boilers, fuel and purchase of outside power. At the same time knowledge and experience on the part of the engineer will be found a valuable asset in either case, as the fuel and repair bill is apt to be a greater expense than is the extra investment in management.

Spare boilers, which are unnecessary in school work, are installed in many cases; and no matter how many are installed, all are likely to be operated to make easier work, thus increasing the repair bill and expense attendant on operating extra fires. Due to the general durability of boilers and the frequent

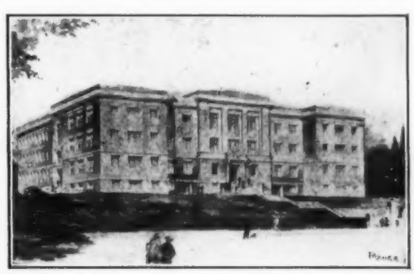

Fig. 1C. Salem High School

intervals between school sessions, any possible breakdown can be handled so as not to interfere with the operation of the plant if the boilers are kept in any kind of general repair. With the wide range in steam requirements for heating, a properly designed system should operate on 50 per cent. of the boiler power required in extreme weather, or 80 per cent. of the time.

Many schools are provided with low-pressure gravity-return systems for the direct radiation, with a separate high-pressure boiler to operate the fan engines, utilizing the exhaust steam in the fan heater. This requires a greater number of boilers than if they were all operated on high pressure and the steam

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taken through a reducing valve; but the use of a pump and receiver during nonsessions is eliminated. Gas engines have also been used to operate fans and do away with operating the boilers under high pressure. All of these schemes have for their object the elimination of intelligent labor at the expense of coal and proper apparatus and do not accomplish the result desired of a lower total operating cost.

The following are the basic methods of arranging the heating system for large buildings in cold climates:

1. The heating proper may be done by low-pressure steam returning to the boiler by gravity or high-pressure steam on the boilers with a reducing valve to lower the pressure of the steam before it enters the heating system. In the latter case a pump and receiver are required to return the condensation to the boilers.

2. Hot-water forced circulation with high-pressure boilers and a live-steam heater with gravity return for all condensation. An exhaust heater is also provided in the water circuit for the use of any exhaust steam from power. These, with two pumps all in series in the water circuit, constitute the hot-water system shown in Fig. 5.

In many schools the radiation is very near the water line of the boilers, thus reducing the operating pressure in many cases to 2 to 5 pounds, so that the condensation will flow from the radiators.

TABLE 1

Outside Temp., Deg. F.	Sessions		Nonsessions	
	%	Hours	%	Hours
0–10	0.6	7	1.6	63
10–20	3.25	36	6.13	241
20–30	13.25	148	17.83	700
30–40	32.4	360	27.82	1,091
40–50	22.7	252	22.42	879
50–60	27.8	309	24.2	950
Totals..	100.00	1,112	100.00	3,924

The low-pressure steam system requires the use of drain traps and expensive trench work to allow a proper gradient, all of which is unnecessary on the water system, due to the pump in the circuit. The mains may be run overhead and exposed as radiating surface, reducing the temperature of the circulated water by that amount. The expensive covering required for steam systems is eliminated except in the boiler and engine rooms.

The live-steam heater arrangement allows any pressure to be carried on the boilers and a gravity return for the condensation is maintained. The boilers may be operated at the same pressure for both power and heating, the exhaust heater utilizing the heat of the exhaust steam.

In all school work the heating load is in excess of any power requirements, and a pressure above atmosphere sufficient to remove the air is always available. With the higher temperature, due to the greater pressure, less surface may be used, and mechanical appliances to produce vacuum and operate the system below atmosphere are unnecessary. A back pressure is not objectionable under these conditions and automatic heat control will reduce the heating steam to a minimum. In making the foregoing statement it is presupposed that the system of piping is properly arranged, as there have been cases where a bad piping layout has been remedied by the use of a vacuum system.

The ventilation and fresh-air supply may be arranged as follows, but one of the above systems is necessary in conjunction:

1. Plenum system with a central stack to heat the outside air to 70 degrees and a supplementary heating stack for air at high temperature distributed through a double-duct system to each flue, with a mixing damper thermostatically controlled.

2. Plenum system with central stack and individual auxiliary stacks at the base of each flue thermostatically controlled.

3. These rules may be modified by concentrating the several individual stacks at the base of each group of flues. A switch damper is provided thermostatically controlled so as to take air at 70 degrees from below the stack or at a higher temperature from above. The thermostat in each room controls the damper in each flue so that as they are turned off the cold rooms have the concentrated power of the entire stack. When the last of the series is taking air at 70 degrees from below, the steam is shut off the entire stack by a relay operating in conjunction with the several thermostats.

4. In a few cases a system of exhaust ventilation only has been provided, openings being made from the outside to individual direct-indirect radiators or to indirect radiators in the basement with heating flues leading to each room. The operation of this system depends on the suction produced by the exhaust fan in the attic to cause a flow of fresh, warm air into the rooms. This system is apt to be unsatisfactory as there is no surety when exhausting air that it will flow from the point desired.

5. Direct radiation operated independently in the rooms with either steam or water circulation for periods of nonsession and a plenum system of air supply furnishing warm air at 70 degrees during school sessions only. The fan may be operated by engine or motor.

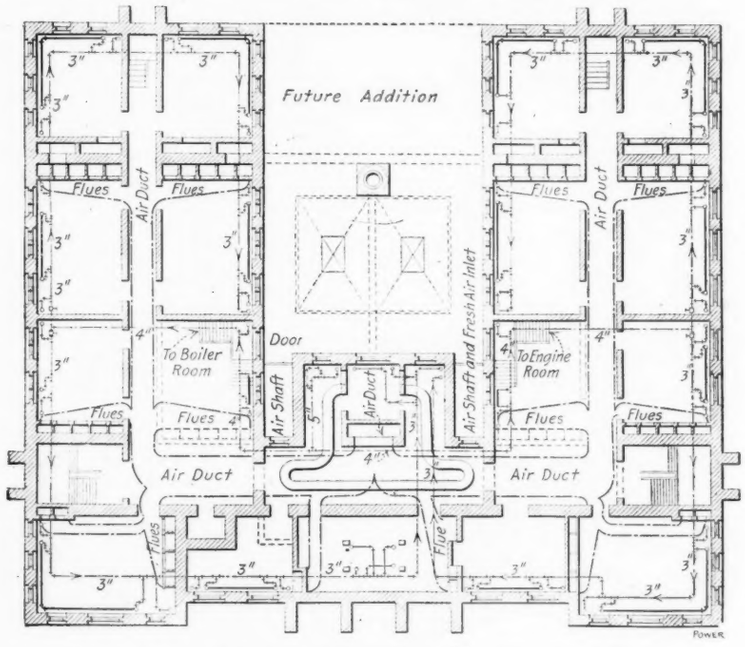


FIG. 2. GENERAL PLAN OF BASEMENT AT BROCKTON SCHOOL

All the above systems except No. 4 may include a system of ventilation composed of vent flues with a fan and motor in the attic or a system of aspirating coils in the flues for exhaust ventilation.

The double-duct system does away with distributing steam pipes to auxiliary stacks, but the hot-air ducts require a greater expense in insulation. The large ducts and the low specific heat of the air make the losses by radiation a serious matter. Mixing dampers are almost impossible to adjust on account of mixing air at different temperatures at various distances from the source of heat and attempting to maintain a constant temperature in the room with the variable mixture of entering air.

The system involving individual stacks is very expensive, due to the large distributing steam pipes and their covering. Traps and trenches for re-

turns are also expensive items. A pump and receiver would also be required, as in most cases, on account of levels, it is impossible to return the condensation to the boilers by gravity.

The modification of combining the auxiliary stacks is open to the same objections except for a reduction in the number of indirect-radiator sections. Indirect radiators are difficult to keep tight where the steam is turned on and off frequently with thermostatic control, due to expansion strains. If very low steam pressure is used, it is sometimes difficult to relieve the stacks of air and fill them with steam quickly. All blower systems require power to operate the fan during nonsession periods, and this may mean a greater steam consumption than is necessary for the heating, especially in moderate weather.

All plenum systems are expensive to operate during nonsession periods, even when provision is made to recirculate the air. The necessity for the system of ventilation for each room causes a large portion of the heated air to escape, and all systems are open to the objection of high steam consumption in extreme weather and radiation losses in transmission. Due to the fresh-air requirements during sessions and the use of entirely outdoor air, there is little difference in the operating expense of all of the systems at that time. The occupants furnish about 300 B.t.u. per hour each in bodily heat, and with 40 or 50 in a room this heat with the fresh-air supply is sufficient to maintain the temperature, except in extreme weather.

There is quite a divergence of opinion as to the positions of inlets and outlets for air in class rooms. It all depends on the density and weight of the air, which varies inversely as its temperature, whether it is foul or fresh. When heating alone is desired the entering air must be warmer and lighter than the air of

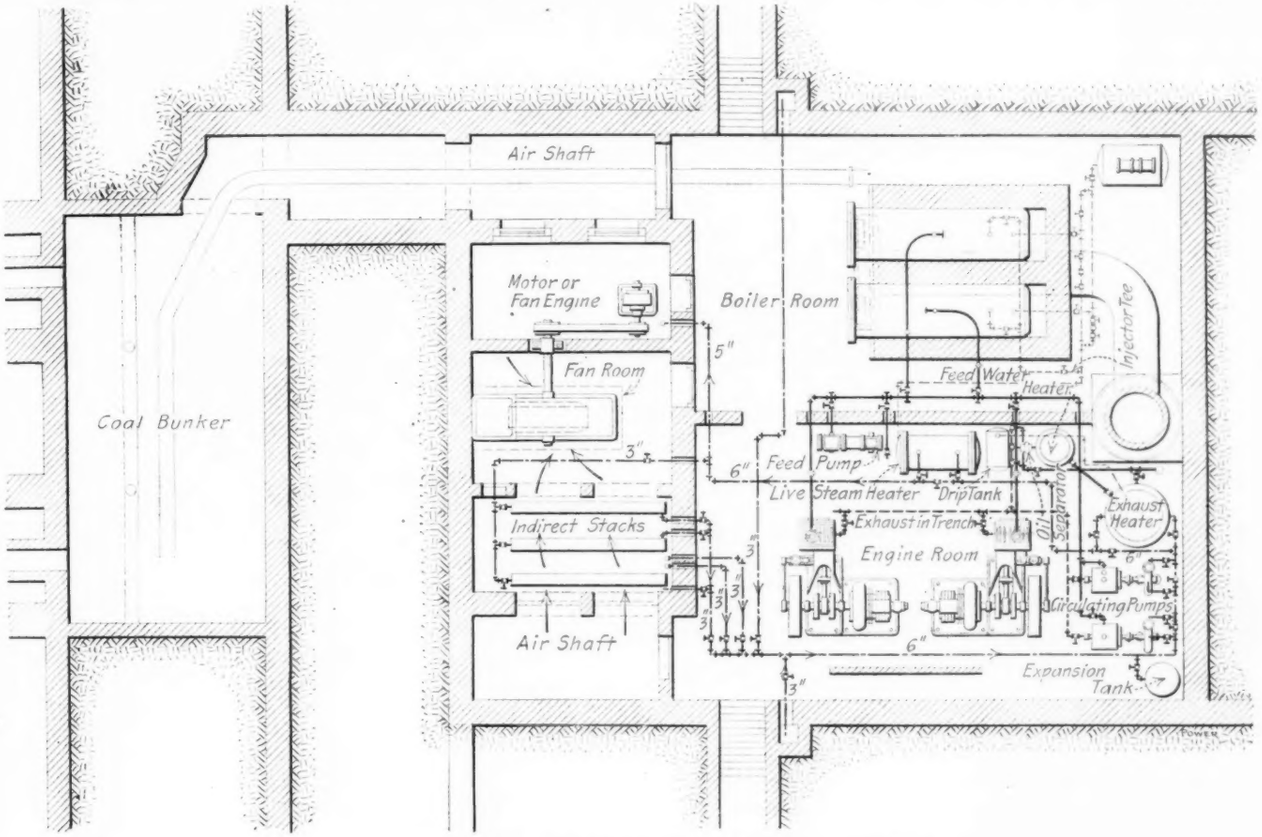


FIG. 3. PLAN OF SUBBASEMENT, SHOWING POWER PLANT

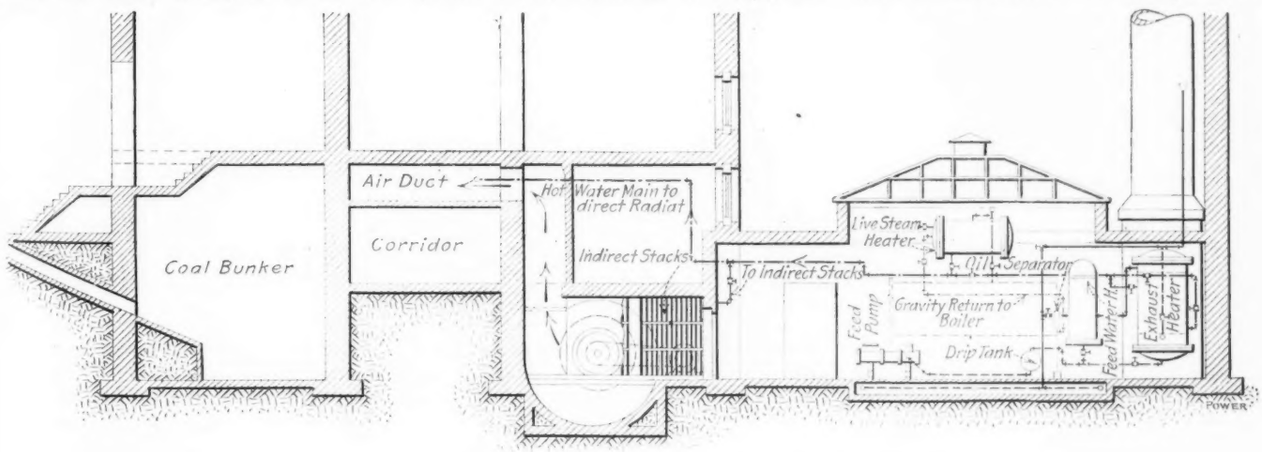


FIG. 4. SECTIONAL ELEVATION THROUGH POWER PLANT

the room and the exit should be at the bottom. When the occupants exhale air and heat is radiated from their bodies, the foul air of any room containing a large number of pupils per cubic contents is lighter than the incoming air and the outlet should be at the ceiling. For a perfect system, both the top and the bottom ventilation should be provided with a register at the ceiling automatically closed when the room is below the temperature of the incoming air and opened when conditions are reversed. This register may be operated in conjunction with the automatic heat control on the direct surface or the auxiliary stack.

The size of the ordinary class room and the frequent air change prevent the temperature changing materially from that of the incoming air and the top register is generally dispensed with in school work without serious results. It is generally agreed that 8 feet from the floor is the proper point for the air inlet.

The foregoing conditions are true for any auditorium holding a large number of people. The bodily heat is sufficient to raise the temperature in conjunction with the necessary volume of entering fresh air which falls and diffuses to a lower temperature without causing drafts. The foul air is then taken through a large ventilator in the ceiling. This condition is reversed when the building is cold and heat is required with no audience present. The larger the building or auditorium the more necessary is top ventilation when people are present.

As to the position of the inlet and outlet with regard to the sides of the room, it makes little difference if the velocities are low. The level and difference in temperature of the contained and entering air form the all important point. The fifth method, being the most economical of operation, will be described most fully.

Fig. 1 shows photographs of three large schools in New England equipped with direct radiation for nonsession heating and a fresh-air supply for periods during sessions, all heated by a system of forced circulation of hot water. The boilers are operated at from 80 to 100 pounds pressure and all the exhaust steam is utilized in the heating system. The South Side high school at Worcester, Mass., which accommodates 900 pupils, requires an air supply of 45,000 cubic feet of air per minute and an 85-horsepower boiler.

The annual report of the superintendent of buildings at Worcester debits this school with $527 for fuel for the year ending November, 1910. The English high school in Worcester, of the same size and operated on steam from a high-pressure boiler used for power, and employing a gravity-return low-pressure system for the heating proper, is debited with $1547 for fuel. Electrical energy from an outside source to operate a 5-horsepower motor on the direct-heating system, nights, was charged at $115 for the South Side school.

This is a difference of $900 in favor of the hot-water school, and at that the motor could easily be done away with by operating the plant nights. The coldest weather known to Worcester is —10 degrees and this school has never been dismissed for lack of proper heating in eight years, which is not true of some others.

As the total cost of fuel from authoritative sources is given and the number of session-hours, it would possibly be of interest to know the cost of operating the fan by outside power. For the school sessions during the heating season, Table 1 gives 1120 hours, and a motor of 28 horsepower capacity, or 21 kilowatts, will furnish power for 45,000 cubic feet of air per minute against ¾ to 1 ounce pressure. Assuming a charge of 4 cents per kilowatt-hour, the cost would be

$$21 \times 4 \times 1120 = 940.80 \text{ } dollars$$

per year, or about twice the cost of the fuel. It is thus apparent that there would be no profit in purchasing power for school work and the saving would pay for good talent to operate the plant at high pressure to supply the engine equipment.

Fig. 1, *C*, is a large high school at Salem, Mass., with about 1500 sittings, which is equipped with two boilers of 110 horsepower capacity each; one only is operated except in very extreme weather. The lowest temperature at this latitude is — 15 degrees Fahrenheit.

Fig. 1, *B*, is the Brockton high school with accommodations for 1500 pupils and the same equipment of boilers as at Salem. The Brockton school has two direct-connected units of 25 kilowatts and 50 kilowatts capacity for lighting and operating motors for the manual-training department. The air supply is 65,000 cubic feet per minute and only one of the 110-horsepower boilers is operated except in extreme weather. The hot-water plant consists of two De Laval centrifugal pumps direct-connected to 15-horsepower turbines of the gearless type, having a capacity of 650 gallons per minute against a 60-foot head; the water connections are in series. The exhaust heater is built of iron with a shell 5 feet in diameter and contains 338 two-inch tubes 6 feet long, which are capable of condensing 4000 pounds of exhaust steam per hour. The live-steam heater is placed vertically in this school and is built of iron with a shell 3 feet 6 inches in diameter with 144 tubes 2 inches in diameter and 5 feet long, capable of condensing 8000 pounds of steam per hour as a maximum. In Figs. 3, 4 and 5 a horizontal heater is shown; this should be of brass tubes of the same capacity on account of the horizontal position requiring provision for expansion.

Fig. 2 shows the general arrangement of the basement of the Brockton school. Due to lack of appropriation, the entire building was not constructed and two front rooms, in the basement, were used for a power plant. These rooms are valuable space and besides the machinery is badly crowded. Fig. 3 shows the arrangement of the subbasement as originally intended before the appropriation was reduced and is applicable to many schools when built on the hollow-square plan. Figs. 3, 4 and 5 show how by excavating 10 feet below the basement-floor level and carrying the main walls of the building down at slight expense, a light boiler and engine room can be had and ample coal-bunker space provided. This gives the use of the basement rooms for school purposes exclusively. It is a vital point in designing an economical plant to have plenty of space for repairing and cleaning the boilers and engines without too much effort.

Fig. 2 shows the air ducts in the corridors, which may be built inexpensively by furring the corridor ceiling, leaving only the branches to be built of galvanized iron. The large coal bunker takes up the dead space behind the front steps. A corridor with tracks connects with the boiler room.

The boiler and engine room is framed with two large skylights between the central and main walls of the building, which will form a light shaft when the additions are completed. There is a passageway around the skylights and two openings are left, one on either side of the projection forming the fan and heater room for light and air. One is used for the fresh-air inlet to the fan, which when the building is completed will be practically taken from the roof level.

In the general plan of boiler and engine room are shown all piping and machinery. The live-steam heater is arranged high on the wall between the boiler and engine room, with a gravity return to the two boilers which are connected to a self-supporting stack for draft. The exhaust pipe is in a trench in the engine-room floor, and the steam passes through an oil separator to the feed-water and exhaust heaters before entering the outboard exhaust pipe. A bypass is also formed around these heaters that the latter may be cut out at any time. The drip from this apparatus drains to a small tank connected with the feed pump. A blowoff tank and sump pump in the rear of the boiler room take care of the blowoff from the boilers and the drainage from the oil separator and oily drips. The automatically operated sump pump empties the blowoff tank to the sewer.

The fan may be operated by motor or engine as desired. The air is discharged through a vertical flue to the ceiling of the basement corridor, as shown in Fig.

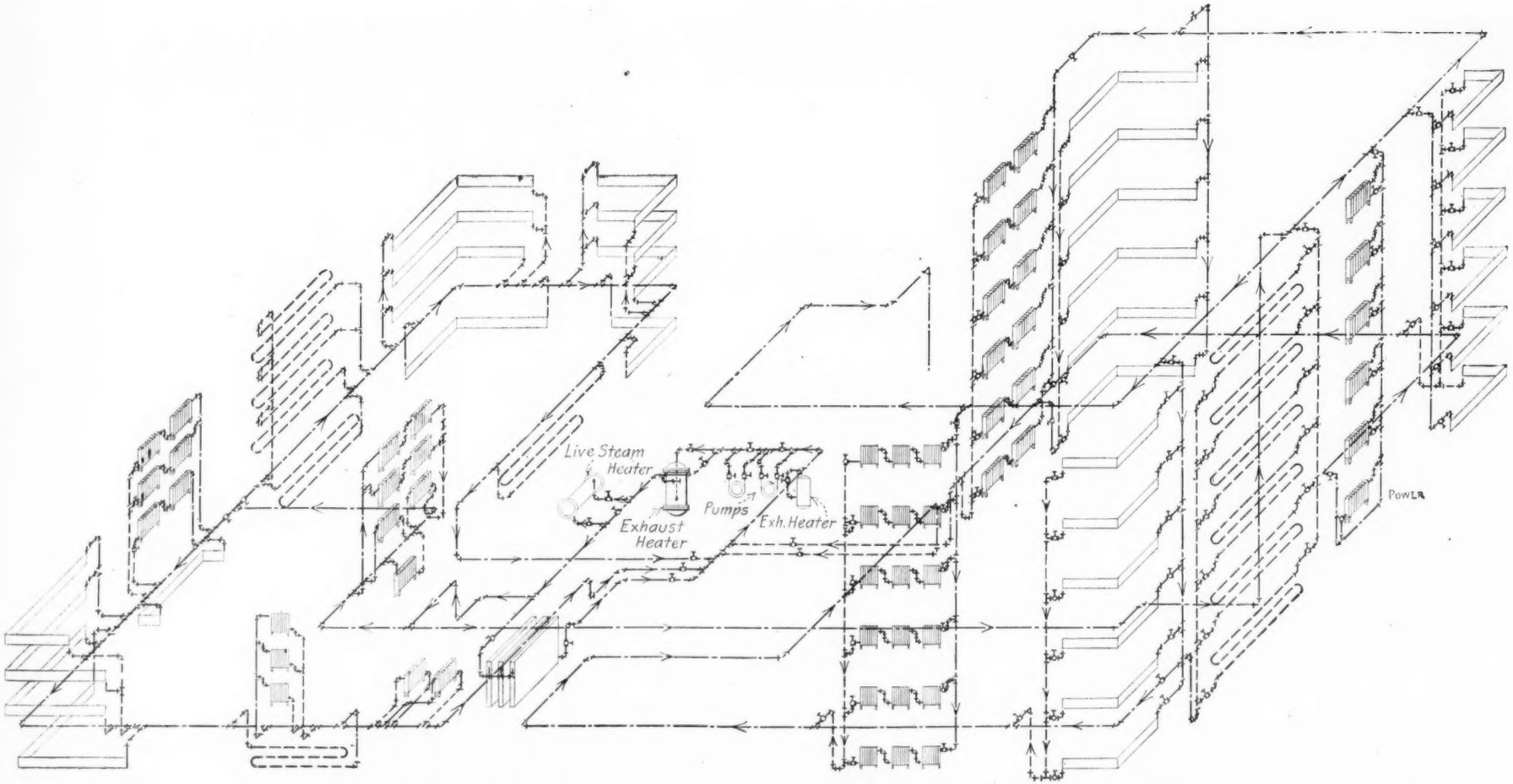


FIG. 5. PIPING CONNECTIONS OF HEATING SYSTEM WITH ONE WING ARRANGED FOR OVERHEAD SYSTEM

4. Three large stacks of cast-iron pin radiators are used to heat the air for the building to 70 degrees or over. They are built up with nipples of 15 square foot sections with an air velocity under 1200 feet per minute. The sections are provided with valves and drawoffs and a separate return connects to the header, as shown in Fig. 3.

If wrought-iron pipe miter coils are used they should be of 1-inch pipe on not more than 2½-inch centers and staggered. For a temperature of 0 to 75 degrees the stack should be 20 pipes deep with a speed of air not greater than 1200 feet per minute.

Fig. 5 shows the arrangement of mains, radiators, coils and risers. The water pipe would not be over 6 inches for the largest school; if the school is not over three stories in hight, the piping should be connected in shunt, as shown. The building is divided in four sections with control valves and thermometers on each return at the header. This makes a single-pipe loop on the basement ceiling not over 3 inches in size.

The second- and third-floor radiators are connected to the supply and return risers and the first-floor radiators have a separate return connection to a 3-inch main, being supplied from the riser. The radiator connections are 1¼ inches throughout and the basement coils or radiators are connected in shunt to the 3-inch main overhead independently.

Valves are unnecessary on the risers in a job of this kind as the four sections of the building are capable of being cut out and the fan system is always avail-

able to supply sufficient heat, if the system of direct radiation gets out of order, until the latter can be repaired.

Fig. 5 also shows one wing arranged with an overhead system which is adapted to a building of over four stories. Supply and return risers are run as shown and coils or radiators may be used for heating surface. The risers in this case should have valves top and bottom with drawoffs. These risers may be reduced to 1¼ inches with 1-inch connections to the coils and radiators. Attention is called to the method of connecting the surface in the basement.

This building has the usual assembly hall and gymnasium, and the fresh-air supply is so arranged that it may be cut out when these places are unoccupied. Direct radiation is provided to keep them warm when not in use.

A ventilating system is provided to remove the foul air with two fans operated by motors in the attic. The toilet rooms and chemical laboratories are ventilated by a separate system on account of the odors when the fan is inoperative.

The expansion tank may be placed in the attic and equipped with a water-feeder pop valve; it is shown in the sketches of the boiler room to be handled by air pressure. This arrangement may be modified to suit any high school. With small pipes and eliminating trenches and expensive covering, the first cost and operating expense should be low. No school should have more than two boilers, and when the horizontal tubular units are too large, water-tube boilers will be found advantageous, as the greater the number of fires the more is the operating expense increased.

Thermostatic control is used on hot water in the same manner as steam, but with the division of circuits the water job can be easily controlled by hand. Thermostats may be arranged to control the flow of water in the four circuits of the building by taking the room of the group supplied most likely to give the best average temperature for the position of the thermostat.

It must be borne in mind that in shutting off a hot-water coil or circuit the heat is not reduced until the water in the coils and radiators cools. Fig. 5 also shows the water connections for the heaters, pumps, expansion tank and the fan coil.

In the Brockton school there is 65,000 cubic feet of air supplied and 14,000 square feet of direct radiation, and there is little difference in the heating during sessions when the air supply is required. A comparison of the nonsession periods will show the saving of one system over another.

In figuring the cost of heating, continuous operation is intended, although many schools, including the South Side school at Worcester, are not operated nights except in extreme weather. There is no saving in fuel by this method and the low-pressure steam. The steam system is expensive in operation, due to the low-pressure boilers and vapor losses. To raise the temperature of 50,000 cubic feet of air one degree, one pound of steam will be required. The transmission of the direct surface will be taken as 1.8 B.t.u. per hour per degree difference in the temperature of the pipes or water and the room. Take the room temperature at 65 degrees for nonsessions and the latent heat of steam at 1000 B.t.u. Then

$$\frac{14{,}000 \times 1.8}{1000} = 25.2 \textit{ pounds}$$

for each degree difference in the temperature of the water and the air of the room. If coal is bought at $4 per ton, of 2000 pounds, and an evaporation of eight pounds is obtained, then the cost for 1000 pounds of steam will be

$$\frac{4 \times 1000}{2000 \times 8} = 0.25 \textit{ or } 25 \textit{ cents}$$

The fan system will require a 50-horsepower engine using approximately 50 pounds of steam per horsepower-hour and 2500 pounds per hour will be the minimum for any period on the fan system. The circulating pump will require 15 horsepower at a maximum of 70 pounds, or 1050 pounds, and this will be the minimum for the hot-water system on direct radiation. This pump must have capacity for the indirect stack which is inoperative at these periods.

For the fan system in zero weather

$$\frac{65{,}000}{50{,}000} \times 60 \times 65 = 5070$$

pounds of steam per hour will be required. Table 2 gives the factors and hours for nonsessions from Table 1, also the requirements for direct radiation on hot water. The system is discontinued at 50 degrees outside temperature.

As shown in the table, the saving of direct radiation over the plenum system is

$$7{,}765{,}000 - 6{,}747{,}000 = 1{,}018{,}000 \textit{ pounds}$$

of steam per season. At 25 cents per 1000 pounds, this means a little over $250, or a saving of 13 per cent. for direct radiation over the fan system.

In a certain large city the janitors are required to hold an engineers' license and they receive a lump sum for the labor involved in cleaning and operating the heating plant; the city furnishes the coal. Some of these buildings contain over a hundred class rooms and require a large boiler plant. Every conceivable scheme is resorted to in order to reduce night operation: gas engines, extra high-pressure boilers and outside power. It is customary to shut the heating plant down nights except what heat can be obtained from banked fires and thus get rid of a night man. In the morning about 20 or 30 pounds pressure is put on and the fan started; 200 horsepower for 10 hours is crowded into about two hours, or into the shortest space of time that the capacity of the installed boiler plant will permit. This naturally results in great waste of coal at $4 per ton and the operation of more boilers than necessary, just to get rid of the $2 to $2.50 wage for a night fireman.

TABLE 2. NONSESSIONS

		Direct Radiation				Plenum System		
Outside Temp., Deg. F.	Hours Non-sessions	Av. Temp. Water, Deg. F.	Dif Temp. between Pipes and Room, Deg. F.	Pounds Steam per Hour	Pounds Steam per Season	Temp. Factors	Pounds Steam per Hour	Pounds Steam per Season
0–10	63	190	125	3,150	198,450	0.85	4,309	271,467
10–20	241	180	115	2,898	698,418	0.65	3,295	794,095
20–30	700	170	105	2,646	1,852,200	0.5	2,535	1,774,500
30–40	1,091	150	85	2,142	2,336,922	fan	2,500	2,727,500
40–50	879	140	75	1,890	1,661,310	engine	2,500	2,197,500
Total...					6,747,300			7,765,062

If a bonus was given the engineer-janitor for each ton of coal saved one season over another, this method of operation would be abandoned, a less number of boilers would be required and the engineer would find that there was a much greater saving to be made in the coal than in the labor.

As it is at present, the city pays ten times as much for coal to operate the schools intermittently as the night labor would cost, and about double the boiler capacity necessary is installed with the attendant cost of maintenance. Hot water would be the favorite system of heating for continuous operation.

Cheaper coal and better loading facilities have led the White Star line to adopt the practice of taking aboard coal in New York for a round trip, and it is quite likely that other transatlantic steamers will adopt the practice. It is said that domestic bunker coal is averaging higher in quality than most of the bunker coals of western Europe and is being supplied at considerable less cost.

Performance of the Field Engine

By L. B. Lent

Description and test results of a steam-air engine built in Scotland in 1894. In spite of an apparently favorable showing the engine never attained prominence.

My attention has been attracted by what has appeared in some of the recent issues of POWER regarding the Christie air-steam engine, in which connection some of the readers may be interested in another air and steam engine which was built and operated about 17 years ago in Scotland.

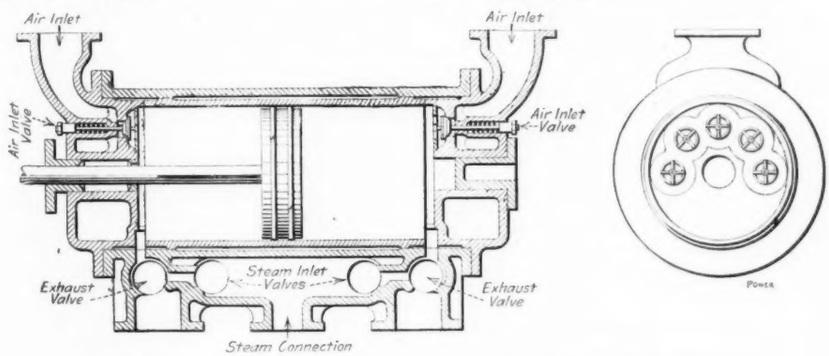
This engine operated on a mixture of steam and air, employing the two-stroke cycle, and was supplied with hot air from an outside source. Test results of this engine are available, and are stated in terms of both indicated and brake horsepower.

A brief description of the engine and of the tests made by Professor Jamieson forms the subject of a paper presented to the Institution of Engineers and Shipbuilders of Scotland, on April 30, 1895. Speaking of the engine under description, Professor Jamieson says:

"This invention is, I understand, the joint design of Edward Field, inventor of Field's well known tubular boiler, F. Saunders Morris, working in conjunction with Musgrave & Co., of Bolton, and George Dixon, their chief engineer.

"It consists of a hot-air pipe connection to the jacket and to each end of a single-cylinder noncondensing engine. A Roots blower, driven by the engine, draws fresh cold air from the engine room, and forces it through a series of heating pipes placed in the main flue between the boilers and the chimney. This heater, therefore, occupies much the same position as a Green's economizer.

"In my experiments, the air was maintained at a mean pressure of 1¾ pounds per square inch, and delivered to the end of the cylinder at a mean temperature of 553 degrees Fahrenheit, and to the valve-casing jacket at about 380 degrees Fahrenheit. This hot air was admitted to the cylinder through special cylinder covers, each containing five inlet valves, which automatically opened inwardly as soon as the exhaust steam commenced to escape. These valves remained open until compression commenced, being held close to their seats by light spiral springs, as shown in the sketch. Consequently, the whole internal surface of the cylinder was heated to a temperature far exceeding that of the steam, thus preventing the possibility of condensation taking place within the cylinder. Under these circumstances, I got the excellent result of 18.6 pounds of steam per indicated horsepower-hour from a single-cylinder noncondensing engine, a result which, as far as I can learn, has never been equaled by any other method of using steam in a single cylinder, and without subsequent condensation."



SECTION THROUGH CYLINDER AND HEAD OF FIELD ENGINE

Without quoting in full Professor Jamieson's description of the specific arrangement of the boiler, heater, blower and engine, and of all the test apparatus, suffice it to say that the described method of test showed that every care was taken to accurately measure the desired quantities and that all instruments, such as feed-water gages, tanks, thermometers, indicators, etc., were carefully calibrated before and after the test, and all measurements were made in an approved manner calculated to insure the greatest accuracy and correctness of results.

Quoting again from the paper: "Steam was kept as steadily as possible at an average pressure of 113 pounds per square inch, and was supplied very dry from the superheater to a Corliss engine made by Musgrave & Co. There was nothing special about this engine, except the application of Field's arrangement; in fact, the engine had not a high mechanical efficiency, since the ratio of its brake horsepower to its indicated horsepower had been found by Professor Kennedy to be only 83 per cent."

No reference is made in the paper to the quality of the steam delivered to the engine other than that stated. However, I am inclined to believe, from an examination of the ordinary Cornish boiler and a knowledge of its construction at that date, that the steam was but slightly superheated, if at all. Since the steam quality is unknown it is impossible to say how much effect it had on the economic results obtained, but probably it was a very small effect.

It is stated that in Professor Kennedy's test of the same engine, under nearly the same load conditions as those existing in the tests under discussion, and running as an ordinary steam engine, the steam consumption was 31 pounds per indicated horsepower-hour.

A summary of the principal results of Professor Jamieson's tests is given in the following table:

SUMMARY OF TESTS

Date of test	November 30, 1894	December 1, 1894
Duration of trials	5 hours	3 hours, 20 minutes
Mean cross-sectional area of cylinder, square inches	280.8	280.8
Length of stroke, feet	2.997	2.997
Mean revolutions per minute	81.66	82.23
Mean boiler pressure, by gage, pounds per square inch	113.4	113.1
Mean initial cylinder pressure by indicator cards, pounds per square inch, gage	113.4	110.7
Terminal pressure by indicator cards, pounds per square inch gage	.....	.....
Mean effective pressure, pounds per square inch	32	13.4
Mean pressure of hot-air supply, pounds per square inch, gage	1.74	1.8
Mean temperature of hot-air supply to inside of cylinder, degrees Fahrenheit	553	530
Mean indicated horsepower	136.75	57.6
Mean brake horsepower	104.6	32.5
Mechanical efficiency, per cent	76.5	67.6
Steam per indicated horsepower-hour, pounds	18.6	21.4
Steam per brake horsepower-hour, pounds	24.26	37.9
Coal per indicated horsepower-hour, pounds	2.45	3.2
Coal per brake horsepower-hour, pounds	3.21	5.68

It is stated that the several alterations which had to be made on the original engine, in order to convert it to the Field's system, were very simple, as they merely consisted in putting on new covers with hot-air inlet valves and hot-air pipe attachments as shown.

Regarding the amount of power absorbed in driving the Root's blower from the engine and heating the air, the quotation from the paper is as follows: "I

understand that Professor Kennedy estimated from his tests of this same engine, both before and after its conversion into Field's system, and under similar conditions of load, etc., that about 5 per cent. of the indicated horsepower was absorbed in driving the Root's blower; but that the heating of the air in the pipes placed in the boiler flue did not appreciably diminish the efficiency of the boilers nor raise the coal bill per unit of water converted into steam. As I had an opportunity only of testing the engine after conversion I accept Professor Kennedy's first results, and, comparing them with my first day's test, I find that the Root's blower lowered the mechanical efficiency of the engine about 6 per cent."

The low steam consumption at low loads (21.4 pounds per indicated horsepower at about ¼ load) is very properly pointed out in the paper; but because of the low mechanical efficiency the steam consumption per brake horsepower is proportionately higher. Discussion of the paper brought out the fact that approximately 20 pounds of air per minute was heated from 60 to 550 degrees and forced into the engine cylinder. This is equivalent to an extraction of 2180 B.t.u. per minute from the stack gases, or to the conversion of less than 2½ pounds of water into steam per minute from and at 212 degrees Fahrenheit.

No inferences can be drawn from the performances of the Field engine regarding the probable performance of the Christie engine, for the cycles on which they operate are very different. The Field engine used hot air delivered to it under pressure and utilized what would be waste heat passing up the boiler stack. The introduction of air at a temperature of 550 degrees Fahrenheit at the commencement of the exhaust stroke certainly must have a beneficial effect in keeping up the metal temperatures. And with an initial temperature of 550 degrees, the temperature at the end of compression must certainly have been such as to superheat steam entering at 114 pounds pressure and thus largely prevent loss by cylinder condensation and other heat losses. And this is all that was claimed for this engine.

It seemes strange, however, that this engine has not, since 1894, attained the prominence to which such a performance should entitle it.

A New Boiler Rating

An interesting and important result of the increasing efficiency of mechanical stokers is the creation of what is practically a new boiler rating. Performance has curiously outstripped theory, and mechanically fired boilers are found which give a horsepower on about 6 square feet of heating surface, over long periods, although boiler manufacturers and the engineering textbooks allow 10 feet per horsepower, or even more.

TABLE 1

Heating Surface per Horsepower, Square Feet	Flue Temperature, Deg. F.
8.6	585
6.8	652
5.8	720
4.9	787
4.3	855

The first purpose of the mechanical stoker was to secure more steady, efficient and economical burning of coal than was easily practicable with hand firing and to cut down the cost of labor. But ability to carry a large overload was a requirement which a stoker soon had to meet if it was to be successful. Then, from carrying a temporary overload to take care of a peak in the power demand, it became practicable for certain stokers to carry an overload of 50 per cent. for practically all the time. This steady performance was equivalent to a new rating for the boilers. Only the gravity underfeed type of stoker can show this performance throughout a reasonably long test, for, although some types can carry an overload for short periods, at the end of about four hours the necessary hand cleaning of the fires seriously cuts down that all-day efficiency. Therefore, the interesting result is that an underfeed stoker, capable of economically burning 60 to 70 pounds of coal per square foot of grate and automatically cleaning fires without checking the combustion or lowering the furnace temperature, has given a new boiler rating of about 6 square feet per horsepower for continuous service.

How far in advance of current theory this is will appear from a comparison of some actual boiler tests with the requirements as laid down in Kent's handbook and as expressed in the rating basis of the best builders.

Kent prescribes 11.25 square feet of heating surface per horsepower, and he quotes the requirements of the American Society of Mechanical Engineers' committee of 1899 to the effect that on this basis a boiler should be able to carry an overload of at least one-third its rated capacity when being pushed regardless of fuel economy.

How far behind performance are these requirements is shown by the two accompanying tables. The first is Kent's estimate of the forced capacity of a series of heating-surface units and the flue-gas temperature which corresponds in each case. It should be noted that with a heating surface of 5.8 square feet, Kent gives a flue temperature of 720 degrees Fahrenheit, which means that the furnace is wasting much fuel up the stack in order to get the high rate of combustion. Kent's figures are shown in Table 1:

Then compare the results given in Table 2 from Taylor stoker tests.

Several important features should be noted. In the first place, Table 2 shows that it is possible for a stoker to carry from 150 to 195 per cent. of rated load for periods which would include several hand cleanings of the fire for any other type. This is the first essential to the ability of a stoker to carry a continuous excess of rating. The next point is that, even with 190 per cent. of rated load, the flue temperature does not exceed 600 degrees Fahrenheit—that is, the waste of fuel up the stack is not seriously detrimental. In several of these cases the flue temperature is noticeably low.

TABLE 2

Plant	Type of Boiler	Heat Surface, per Horsepower	Flue Temperature	Builders' Rating, Per Cent.	Duration of Test, Hours
Narragansett Electric Lighting Co	B. & W.	5.5	544.2	180.4	8
National Museum	Geary W. T.	6.4	430	155	16
Commonwealth Edison Co	B. & W.	4.97	588	201	7.5
Everett Mills	Manning	6	599	150	39
N. Y. Edison Waterside	B. & W.	5.48	550	179	8
Old Colony Street Railway	B. & W.	5.25	599	190	12
West Albany Shops	Franklin and Edge Moor	5.28	543	193	14
New York Central Railroad					
Cambria Steel Co	B. & W.	5.14	485	194.5	9

Boiler Inspectors Meet at Boston

The second semi-annual meeting and banquet of the American Institute of Steam Boiler Inspectors was held at the American house, Boston, Mass., on Tuesday evening, October 24, and was largely attended.

Arrangements were made by the educational committee for outside talent to speak, it securing the services of H. M. Feldman, representing the Chandler & Floyd Company, who gave a very interesting talk on the manufacture of Pittsburg "pure brand" American ingot iron, manufactured by the Allegany Steel Company. A. M. Lloyd, representing the Central Iron and Steel Company, gave a very interesting as well as humorous talk. Frank S. Allen, chief inspector for the Hartford Steam Boiler Inspection and Insurance Company, was the next speaker, and the relation of his experiences during 40 years of steam-boiler inspection was immensly enjoyed.

The American Institute of Steam Boiler Inspectors is an educational and social organization and has had a very rapid growth. It has among its members, boiler inspectors from all parts of the United States, Canada and Alaska. Thomas G. Ranton, 112 Water street, Boston, Mass., is secretary.

Engine Badly Wrecked

By E. B. Emerson

On the morning of October 8, at the the plant of the Canton (Ohio) Sheet Steel Company, a 20x36-inch Corliss engine running at a speed of 100 revolutions per minute was completely wrecked. The engine was used to drive a 400-kilowatt alternating-current generator.

Steam was furnished by four water-tube boilers. At the time of the wreck there was only one boiler in service, it being the one farthest away from the engine. The boilers were connected to a 14-inch header and the wrecked engine took its steam from the end of the header through an ell, reducer and 7-inch pipe which in a half-circle bend dropped to the steam chest 10 feet below the level of the header. The header had a slight fall toward the engine and was not provided with a separator, trap or other means of draining the condensing water. It being Sunday, only the engine was drawing steam from the header, and it would appear that enough condensation went over into the cylinder to cause all the trouble.

The steam chest was cracked from end to end. The wristplate stud was broken off, the piston rod pulled out of the piston head and the connecting rod was bent. The cap of the main bearing was broken into two parts, the main-bearing pedestal was split in half and broken off, the outboard box was broken and the eccentric cracked. The reach rod and all of the steam- and exhaust-valve rods and the trip collar and dashpot rods were bent and there were several other minor cracks and bends.

The entire trouble could have been avoided by the installation of a separator at the engine. Also, the header should have been properly drained.

OBITUARY

As briefly announced in Power of October 31, Robert Mather, chairman of the board of directors of the Westinghouse Electric and Manufacturing Company, died at his home in New York City, on October 24, of acute peritonitis.

Mr. Mather was a fine example of that group of American railroad men who began their business lives in the shops and, thus equipped with a thorough working knowledge of mechanics, rose by their often unaided efforts and keen grasp on opportunities to positions of great responsibility.

Born at Salt Lake City in 1859, Robert Mather was educated in the Galesburg (Ill.) public schools, leaving the high school at the age of 13 because he had to earn his own living. For three years he was learning the manufacture of telegraph and switchboard apparatus, when he secured a position in the master mechanic's office of the Chicago, Burlington & Quincy Railroad at its Galesburg shops. Having meanwhile prepared himself for college in his spare hours, he entered the freshman class of Knox College in 1877. His money being exhausted at the end of his first year, young Mather resumed his work in the railroad office. Reëntering Knox, he graduated in 1882

Robert Mather

with the degree of A. B.; in 1885 he received his A. M., and in 1907 the honorary degree of LL. D.

Having been previously admitted to the bar, Mr. Mather, in 1889, became local attorney and from 1894 to 1902 general attorney of the Chicago, Burlington & Quincy Railroad.

He held high positions in the executive boards and directorships in several of the largest railroads in the country.

In January, 1909, Mr. Mather was made chairman of the board of directors of the Westinghouse Electric and Manufacturing Company, and thereupon severed most of his railroad connections. He was also a director of the Equitable Life Assurance Society of the United States, the Mercantile Trust Company, the Havana Electric Railway Company, the Westinghouse Lamp Company, the Canadian Westinghouse Company, the R. D. Nuttall Company, the Niagara, Lockport & Ontario Power Company, the Perkins Electric Switch Manufacturing Company, the Bryant Electric Company, the National Bank of the Republic, of Chicago; the Chicago, Rock Island & El Paso Railway Company and general counsel of the Chicago & Alton Railroad Company.

Mr. Mather was married in Detroit in 1892 to Alice Caroline, a daughter of Horatio Jell, of Walkerville, Canada.

Isolated Plant Victory

It is said that the Henry Siegel Company, which has been using the Edison service in its department stores upon Washington street, Boston, will, as soon as its present contract expires, put in its own plant, and has already commenced to accumulate apparatus for that purpose.

PERSONAL

William Scott Taggart, member of the German Society of Mechanical Engineers and the Textile Institute, has opened offices at 22 Bridge street, Manchester, England, as a consulting engineer in general-engineering and textile work.

C. D. Chasteney has resigned his position as sales manager of the De Laval Steam Turbine Company, of Trenton, N. J., having acquired an interest in the Turbine Equipment Company, of 30 Church street, New York, which company represents the De Laval Steam Turbine Company in New York State, parts of New Jersey and Connecticut. Mr. Chasteney was graduated from Stevens Institute of Technology in 1901 and has been with the De Laval Steam Turbine Company since the organization of the American company, over 10 years ago.

SOCIETY NOTES

In connection with the New England textile exhibition, to be held in the building of the Massachusetts Charitable Mechanics Association at Boston, commencing April 22, it is proposed to run a power section under the auspices of the New England Association of Commercial Engineers.

The largest and most successful entertainment and reception yet held by the Universal Craftsmen Council of Engineers, of Manhattan and vicinity, took place on Friday evening, October 27, at Lexington opera house, New York City. An interesting vaudeville performance was followed by a long dance program. There were present many prominent persons in the field of operating engineering.

The annual banquet of the Engineers' Blue Club, of Jersey City, N. J., was held at Columbian hall, on Saturday evening, October 28. Seated at the tables were about two hundred members and guests. John J. Calahan was the toastmaster and introduced the following speakers: F. L. Johnson, James R. Coe, Charles F. X. O'Brien, Patrick Flannery, Mayor H. O. Wittpenn, Hon. Robert Carey, George F. Tenant, president of the board of education, Edward A. Murphy, Ph.M., John H. Foote and Frank Broaker. Between the speeches members of the New York "Bunch" entertained.

Moments with the Ad. Editor

A department for subscribers edited by the advertising service department of Power

There is a further side to this relation of the reader to the ads that wasn't touched on by the man who wrote the letter on the scrap-book idea the other day.

"It is very interesting," says another man, "if you have a file of the paper to compare the ads of ——— firm 20 years ago and see the difference between the machine made by it then and now."

This idea of studying the development and evolution of a machine by following the ads through successive years, is a mighty interesting one. It is only another way of following the growth of the firm itself and observing how *it* has developed by adoption of the latest methods and has worked step by step to its present position. One can see, in an ad. study like that, how one firm has outrun another in the race for higher efficiency, and again how a certain firm came up 20 or 30 years ago with a product so good that it has led the field ever since with few changes. One can see how new firms have started up with new products, products that have done away with losses and exasperating troubles in the power plant which you, maybe 20 years ago, thought you would have to put up with till the end of time. But always *the* interesting thing in that study of firms' growth through the ads is to see who has *gone ahead*. Being a little better today than yesterday, and a little better yesterday than the day before, is the real winner of belief in anything—man, firm or machine.

There is another side to this comparison, though, quite as important. It's the difference in the ads themselves.

What this is you can notice at once by putting a page of 20-year-ago ads beside a page of today's.

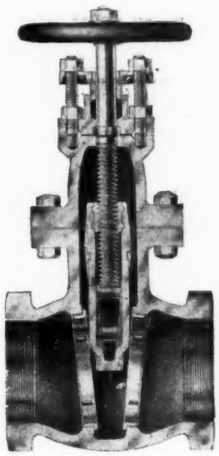
Then the firms advertised their own names. Now they advertise their goods.

Selling by advertising the firm name only was and is a left-over from the day when a proclamation that a firm was "Caterer to the King" was enough to sell its goods, no matter what the quality of the goods dished out by that same concern to the people who weren't kings.

But today most firms talk about their goods, not themselves. And this is because there is more and more pressure on what the machine, the boiler, the lubricant, the valve themselves can do—no matter who makes them.

All that change in the way of writing ads is due to a study of *you*, the reader and buyer. The association with the quick, vigorous force of the up-todate power-plant paper has put you on the alert for *things full of ideas*. Well, the only ad. worthy to be coupled with such a thing is itself *full of ideas*—it's the only kind of ad. that will get your eye.

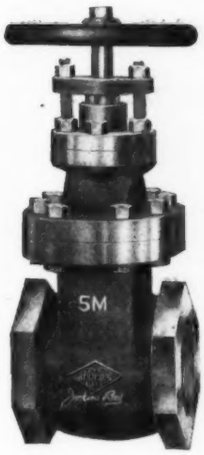
Look through POWER'S Selling Section, and see.



TRADE JENKINS MARK

Jenkins Bros.

Luck Doesn't Produce A Good Valve



Some engineers when they get hold of a valve that's really efficient consider themselves lucky.

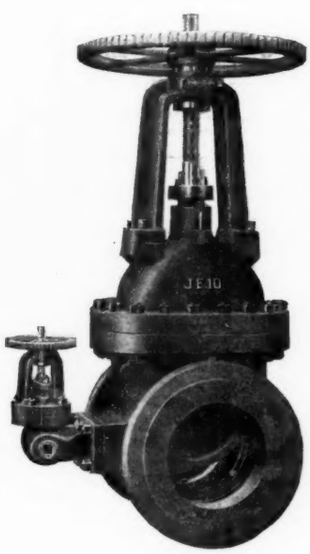
As a matter-of-fact there's absolutely no luck connected with the making of a good valve.

It's all hard, clear and concise knowledge and years of experience,—it's the employment of the best material used in the best way that enables us to make Jenkins Bros. Valves as good as they are. Take the valves illustrated:

Jenkins Bros. Medium Pressure Gate Valves are tested to 500 lbs. hydraulic pressure and guaranteed for 175 lbs. working pressure.

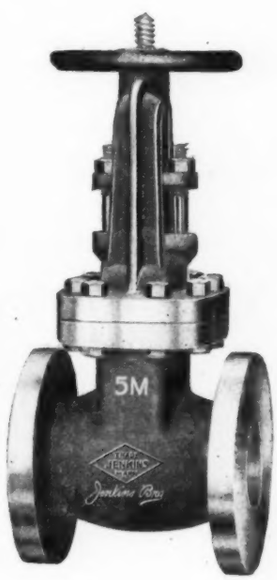
The Heavy Pressure Gate Valves are guaranteed to the most severe high pressure steam or water service.

When you buy Jenkins Bros. Valves you eliminate luck and in its place get certainty—the certainty of getting a thoroughly reliable, dependable and efficient valve *every time*.



Ask your dealer for Jenkins Bros. products. Insist on getting the *genuine*. If he fails to supply you, drop us a line for name of nearest dealer who will.

We'll be glad to send you a copy of our latest catalog.



Jenkins Bros.

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35 High St., Boston
133 N. 7th St., Philadelphia
300 West Lake St., Chicago

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Armstrong Mfg. Co.....102
Curtis & Curtis Co.....103
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Williams & Co., J. H.....126

YOU Can Go Where Others Go

The only difficult part of GOING UP is in making up your mind to START. After that, the road is smooth, and the higher you go the easier it becomes. You may be working under such circumstances that advancement seems impossible, but YOU CAN GO UP. You may now have a fairly good position, but you can go still higher. Just follow the example set by thousands of I. C. S. students who have made good and are making good. Every month an average of over 400 students of the International Correspondence Schools **voluntarily** report an increase in their earnings.

The I. C. S. actually takes the workingman by the hand and helps him to prosperity.

Think of a man who a short time ago was earning but $10 a week and is now earning five times that amount.

Think of a day laborer being qualified as superintendent by I. C. S. training.

These are not exceptional cases. There are thousands and thousands of them. The I. C. S. will tell you who they are and where they are.

Architects, Engineers, Electricians, Designers, Draftsmen, Inventors, have been and are students of the I. C. S. Large employers of labor regard I. C. S. training as proof of ability.

Do you doubt its power to help you? If you do, here is an offer to prove it. Mark and mail the coupon at once, and the I. C. S. will send you more evidence. If you are not perfectly satisfied you are under no obligation to proceed further.

International Correspondence Schools

Box 979, SCRANTON, PA.

Please explain, without further obligation on my part, how I can qualify for a larger salary and advancement to the position, trade, or profession before which I have marked X.

Electrical Engineer	Machinist	Chemist
Electrical Mach. Des.	Toolmaking	Assayer
Dynamo Foreman	Molding	Commer'l Illustrat'g
Electric Lighting	Blacksmithing	Bookkeeper
Electric Railways	Civil Engineer	Stenographer
Electrician	Stationary Engineer	Civil Service Exams.
Telephone Expert	Gas Engineer	Commercial Law
Concrete Construct'n	Refrigeration Eng.	Architecture
Mechanical Engineer	Sheet-Metal Drafts.	Contracting & B'ld'g
Machine Designer	Marine Engineer	Advertising Man
Mechanical Draft.	Mining Engineer	Window Trimming
Patternmaking	Structural Engineer	Automobile Running

Name _____

Street and No. _____

City _____ *State* _____

Present Occupation _____